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ADVISORY GROUP FOR AEROSPACE RESEARCH & DEVELOPMENT

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AGARD LECTURE SERIES No. 83

on

Modern Prediction Methods for Turbomachine Performance

NORTH ATLANTIC TREATY ORGANIZATION

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NORTH ATLANTIC TREATY ORGANIZATION

ADVISORY GROUP FOR AEROSPACE RESEARCH AND DEVELOPMENT

(ORGANISATION DU TRAITE DE L'ATLANTIQUE NORD)

AGARD Lecture Series No.83

MODERN PREDICTION METHODS FOR TURBOMACHINE PERFORMANCE

The material in this book has been assembled in support of a Lecture Series presented in Munich, Germany on 14–15 June 1976 and in London on 17–18 June 1976, sponsored by the Propulsion and Energetics Panel and organised by the Consultant and Exchange Programme of AGARD.

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Published June 1976

ISBN 92-835-1219-7

621–135:621.438

Printed by Technical Editing and Reproduction Ltd
Harford House, 7–9 Charlotte St, London, W1P 1HD

PREFACE

This Lecture Series, No.83 on the subject of Modern Prediction Methods for Turbomachine Performance, is sponsored by the Propulsion and Energetics Panel of AGARD and implemented by the Consultant and Exchange Programme.

Propulsion system development costs may be significantly reduced by improvement of methods for prediction of compressor and turbine component performance, and by preliminary study of the interactive operation of compressors and turbines with other system components. After the build-up of development engines, it is necessary to understand and carefully plan the process of rematching of components for optimum system performance.

AGARD Lecture Series Number 83 includes lectures and a panel discussion on the historical background of turbomachine performance prediction, on current procedures for estimation of overall and blade row performance characteristics, and on qualitative and quantitative turbomachine performance information needed for evaluation of the effects of compressor and turbine behaviour on the complete propulsion system. The lectures on component performance prediction cover both current and developing technology for axial-flow compressors and turbines, centrifugal compressors and radial-inflow turbines.

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CONTENTS

INTRODUCTION

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Aircraft propulsion system development time and cost could be significantly reduced if we could predict, with precision and confidence, the performance characteristics of fan, compressor and turbine components. This comment, in different forms and in different languages has been repeated on countless occasions during the past thirty years in meetings and in publications of technical societies and AGARD, as well as in the internal discussions and program plans of government agencies, manufacturers and academic research institutes. The comment is "understood" by engineers and scientists working in all phases of turbojet and turbofan engine and component research and development. It is, however, necessary to be certain that the comment and the real nature of the problem are not only "understood" by these engineers and scientists, but also that they "understand" each other qualitatively and quantitatively when they discuss it. We hope that this Lecture Series will bring together the turbomachinery specialists with those working at all levels of the aircraft engine development process. We hope to define carefully requirements, methods and currently achievable results.

Prediction of turbomachine performance is one of the most difficult problems in applied fluid dynamics. Flow patterns in compressors and turbines operating at their design points are complex, but operation at off-design flows and rotational speeds, or with nondesign entrance flow distributions or exit boundary conditions introduces additional regions of separated or choked flow, often accompanied by excessive aerodynamic or aeromechanical instability. Furthermore, all of these difficulties occur at so-called "equilibrium" turbomachine operating points. The time-dependent behavior of a fan, compressor or turbine component when operating in the engine environment, with heat transfer and storage plus gas dynamic phenomena, adds another disturbing dimension to the story.

In the first lecture Dr.-Ing. K. Bauerfeind will describe turbomachine performance prediction requirements from the point of view of propulsion and flight system simulation and performance prediction. I believe that his review of the current status of simulation programs and capabilities will establish objectives for evaluating our present position in estimating and computing turbomachine performance.

My lecture will then follow the pattern of development of compressor and turbine performance prediction systems during the past thirty years. I will try to tell you what these systems are and what they have done in the past. Each group of prediction systems has limitations which should not be disguised or neglected.

Mr. R. A. Wall will then describe the employment of performance prediction techniques in the process of design selection and optimization of axial-flow fan and compressor units for engine application. His presentation will emphasize the present situation in compressor performance estimation and some ideas which may influence future component designs.

In the fourth lecture R. A. Novak has summarized recent progress in his long-standing efforts in computation of turbomachine internal flow fields. In this most difficult area of performance prediction he has given us through the years some of the best available computation systems and results for typical compressor and turbine configurations.

Next Dr. David Japikse will review the situation in performance optimization and estimation for centrifugal compressors and radial-inflow turbines. Unique features of the overall problem exist for these geometries and they will be discussed as we attempt to find common bases for our future work.

M. A. G. Habrard will deliver the last formal lecture as a description of how predicted and measured installed performance of turbomachines are utilized in the more advanced phases of the propulsion engine development process. Requirements and methods used at this stage are somewhat different and call for integration and evaluation of immense quantities of information in order to achieve optimum engine performance.

To conclude the Lecture Series, we will all serve as members of a "round table" or panel, to discuss conclusions and recommendations. I would like to request full and active participation of everyone in attendance during this discussion. We have an extensive and important subject to review and your suggestions and ideas will be welcome.

AIRCRAFT GAS TURBINE CYCLE PROGRAMS - REQUIREMENTS FOR COMPRESSOR AND TURBINE PERFORMANCE PREDICTION

Dr.-Ing. K. Bauerfeind
Motoren- und Turbinen-Union München GmbH

SUMMARY

Over the past two decades a lot of effort has been devoted to the development of engine computer models, since the success of an engine program depends to a large extent on the quality of the thermodynamic engine model. The simulated engine performance data is the basis of practically all the engine design work. Furthermore it is being used for contractual work as well as for the evaluation of the flight performance of the aircraft. Simulations are carried out to understand and solve development problems and to investigate failure cases. For a flight simulator the engine model represents a very essential part. Such a modern model of a gas turbine engine does not only cover the steady state working range between idle speed and full power but can also simulate starting, windmilling and transient performance respectively around the flight envelope. It is quite obvious that the accuracy of the performance predictions will primarily depend on the quality of the component performance used. In this context both the compressor and turbine characteristics are of particular interest. The turbo machine specialists are required to predict and specify the performance of their components in a suitable form over a very wide working range. They must also consider and specify the installation effects for their components when working in a practical engine with different clearances and in some cases suffering from aerodynamic interaction effects with other components.

CONTENTS

1. TYPICAL APPLICATIONS FOR AN ENGINE PERFORMANCE PROGRAM

A comprehensive engine performance program can generally be used in two different modes, namely the engine design mode and the synthesis mode. During the initial phase of a new engine project it is being operated as a tool to optimize the thermodynamic cycle data to obtain the best possible match of the many prevailing requirements. This optimisation work is in many cases a complex iterative process. Usually the performance engineer starts with a rough guess of the component performance and scales these components in a suitable manner in order to come as close as possible to the performance requirements for his new engine. The component performance engineers have then to refine the assumptions made on the basis of the technology available. Together with the inputs from design, stress and cost evaluation the new engine takes shape and the cycle data together with the main dimensions of the components and flow passages are fixed.

After the engine has been aero-thermodynamically defined and the important control limits specified the second mode of the engine performance program can be used to compute installed and uninstalled engine performance around the flight envelope for different power settings, air and power off-takes etc. The engine performance program is now being used to produce performance data required by the airframe manufacturer, to establish performance guarantees, to provide the information required to stress the engine and to specify the subsystems to name just a few of the main objectives at the beginning of a project. Immediately afterwards information is required about things like start up performance, windmilling and altitude light up capability, rapid thrust changes etc. The simulation of failure cases, production tolerances etc. is yet another area where the performance program is being used.

It is clear that there is a definite requirement to continuously update the engine model with the best and most realistic information available at the time. This information can come from either rig or actual engine testing. In most cases it is practical to have at least two engine models available, one for the development standard achieved at the time and the other one for the target standard. Fig. 1 gives a very rough indication of the important milestones during an engine development program, at which points of time updated component information is required.

2. CHARACTERISTIC STRUCTURE OF A THERMODYNAMIC ENGINE MODEL FOR STEADY STATE PERFORMANCE

Fig. 2 shows in very general terms the structure of such an engine computer model. The computation routines can be controlled from the input to either work in mode 1 (specification of the cycle data - determination of the geometry) or in mode 2 (specification of the engine geometry - computation of the cycle data). The computations required for a single shaft straight jet engine according to mode 1 are a fairly simple and straight forward affair. On the other hand the computation of the performance (mode 2) of a three-spool bypass engine with mixed streams and afterburning is already a very lengthly and complex job. To compute just one operating point would take a man equipped with a desk calculating machine between 1 and 2 months. Therefore very fast electronic digital computers are being used. To give an example for the already mentioned three spool bypass engine with mixed streams and afterburning, MTU uses an IBM 370/155. A storage capacity of 250 K is required and the computing time is approx. 5 sec. It can be shown that the type of computation routines used is very important as far as the required computing time is concerned since a lot of iterations have to be performed. The aero engine manufacturers usually invest a lot of money into the development of their engine computer programs.

3. NON-DIMENSIONAL ENGINE PERFORMANCE DERIVED FROM NON-DIMENSIONAL COMPONENT PERFORMANCE

The characteristics of the turbomachines are usually presented in terms of parameters which are equivalent to two characteristic Mach-Numbers. It should be noted that in some cases the so called non-dimensional parameters are in actual fact dimensional because certain constants have been omitted for reasons of convenience. Fig. 3 shows a typical compressor characteristic, where the inlet mass flow parameter (abscissa) is equivalent to an axial Mach-Number, while the speed parameter $N/\sqrt{T_t}$ represents a Mach-Number in circumferential direction. Instead of the total pressure ratio sometimes the parameter H/T_t is being used. The efficiencies can be defined as either isentropic or polytropic. Like with most components, compressors have certain operational limits beyond which operation is not possible. These limits are:

- Physical shaft speed N_{max}
- Non-dimensional speed $(N/\sqrt{T_t})_{max}$
- Choke flutter boundary at low operating points where blades reach a dangerous vibration level due to high local Mach-Numbers (only with some compressors)
- Surge line above which one blade row stalls leading to flow and pressure fluctuations which can be mild at low speeds but can be dangerous to the mechanical integrity of the component when prevailing for a longer period at high speeds.

It is very essential to establish these limits as soon as possible and as precisely as possible. With respect to the surge margin required from a compressor for a particular job the situation is rather complex. Based on his simulation results the performance engineer can define the surge margin required in the engine. This requirement has then to be translated into the design or rig surge margin requirement, depending on the flow distortion pattern the compressor has to cope with in the engine and on its own sensitivity to these distortions. This distortion pattern, however, depends to a large degree on the aircraft intake characteristic and on the distortion attenuation or generation characteristic respectively of the compressor operating upstream in the case of a multi compressor arrangement. Depending on the design rules used for the layout of a particular fan or compressor, certain types of distortion patterns prevailing in the engine can actually improve its surge line relative to the rig surge line. At least one engine is known where a compressor is successfully operating in the installed engine with a negative rig surge margin due to a favourable radial pressure profile. In most cases, however, some rig surge margin is lost in the engine relative to the rig. This is particularly so with compressors working behind the fan in the core engine. Here the loss of surge margin can be excessive if no special counter measures are being taken.

Fig. 4 shows the equivalent characteristic for a turbine. Again the mass flow parameter $M\sqrt{T_t}/P_t$ is representative of an axial and the parameter $N/\sqrt{T_t}$ of a circumferential Mach-Number. Apart from the mechanical and thermal limits in terms of $N_{H\,max}$ and maximum permissable inlet temperature the turbine usually presents much less of a problem with respect to aerodynamic limits. This is mainly so because it is always easier to accelerate a gas flow rather than to decelerate it as in compressors. Apart from this, turbines usually work only over a very limited range of their characteristic under normal engine operating conditions. Exceptions are engine start ups and emergency shut downs and certain failure cases.

After having defined the characteristics of all the components the computation of engine performance can start. Although the thermodynamic basis of the computation itself is fairly straight forward the required complexity of such a computer program can become almost frightening when dealing for instance with three-spool bypass engines with afterburning and complex secondary air systems. The normal approach is always to guess certain parameters at the beginning like compressor operating points etc. and to check later in the computation process whether the fundamental conditions like equal work between compressors and turbines, preservation of mass etc. are being met. In case of disagreement an iteration on the original guesses is carried out until the error is within a specified bracket.

Provided the assumption is made that:

- the flow through the engine is truely one-dimensional and
- the turbomachines follow their characteristics over a wide range of absolute pressures and temperatures

then all the thermodynamic data can be expressed in terms of non-dimensional parameters like pressure and temperature ratios, mass flow and fuel flow parameters etc.

4. CORRECTIONS FOR TYPICAL DIMENSIONAL EFFECTS

The presentation of engine performance in terms of non-dimensional performance parameters had been the standard method for many years after the introduction of the jet engine. With an extension of the operational range of both military as well as civil aircraft, the shortcomings of this method became more and more apparent. The main reason for the discrepancies between predicted non-dimensional performance and actually measured performance lies in the fact that the engine components and here mainly the turbomachines can change their non-dimensional performance consideralby when working in an engine over a wide range of inlet conditions. There are a number of basic influences on the compressor and turbine performance respectively which will be briefly dealt with on a qualitative basis. For a comprehensive engine per-

formance computer model it is absolutely essential to quantify these influences as precisely as possible. These influences can affect efficiency, mass flow and surge margin in the compressor and efficiency and mass flow in the turbine. A deterioration of the efficiencies will always lead to a reduction of thrust and an increase of the specific fuel consumption. A reduction of the mass flow at a speed, however, does not necessarily lead to any performance degradation at all providing the shaft speeds are still clear of their respective limits. A drop of the compressor surge line on the other hand adversely affects the handling qualities of an engine by reducing the potential for both fast accelerations and power off-takes respectively. Depending on the particular case the degradation of a rotating component relative to its ideal rig performance can be appreciable.

4.1 REYNOLDS-NUMBER CHANGES

It has been known for a long time that the non-dimensional performance of turbo machines can be affected when reducing its Reynolds-Number below a certain level The critical level below which this influence becomes important depends on the design parameters of the component, including things like blade surface quality etc. When operating in this regime an increasing deterioration of the component performance in terms of efficiency, mass flow and compressor surge line can be observed. The level and range of Reynolds-Numbers encountered in an engine depend on the dimensions of the particular component and its working pressures and temperatures respectively. In many cases it has proven sufficient to present the Reynolds-Number effect in a simplified form as shown in Fig. 5, where the temperature term has been neglected. There is usually a definite requirement to rig test compressors and turbines which have to work below the critical Reynolds-Number level over an as wide as possible range of absolute pressures. This usually requires testing in a high altitude test facility in order to simulate at least something like the actual inlet conditions as encountered in the engine.

4.2 CLEARANCE CHANGES

It is the usual aim for any good design to keep the changes of tip and seal clearances over the engine range of operation as small as possible. A lot of effort has been devoted to the development of compensating casings, abraidable coatings etc. Nevertheless, after the running in procedure certain clearances do exist and they usually change over the operating range as a function of the temperature distribution and rotational speeds respectively. It is therefore essential to explore on the rig the influence of different clearances on the component mass flow, efficiency and surge margin and to compute the realistic clearances at least for a number of important corner points within the operational and flight envelope of the engine.

4.3 PRESSURE AND TEMPERATURE PROFILES

Originally all engine performance calculations had been based on the assumption of one-dimensional flow with constant pressures and temperatures prevailing at each cross section of the flow paths of the engine. Soon it became apparent, however, that this assumption is not justified in many applications. The aircraft intake togehter with the angle of attack or sideslip of the aircraft can produce rather severe pressure distortions of both the static and the instantaneous type in the entry plane of the first compressor. These distortions are then being more or less attenuated in the downstream components of the engine. Apart from the intake/flight anlge induced distortions some of the engine components themselves can produce considerable distortions. Typical examples are the fan of a bypass engine whose flow split usually varies with operating conditions, resulting in sometimes considerable pressure and temperature gradients in its outlet plane. Experience has shown that at least for the low bypass ratio engine realistic fan performance can be presented by defining exit temperature and pressure profiles varying as a function of $N/\sqrt{T_t}$ and/or bypass ratio. With the high bypass ratio fan usually a number of characteristics is being generated for distinct bypass ratios. These characteristics are stored in the computer which can also interpolate between the characteristics.

As mentioned already there is usually a pronounced aerodynamic interaction effect between the fan and the following core engine compressor with the main effect on surge margins. Support struts, swan necks etc. are components

which induce circumferential and radial profiles respectively. The combustion chamber on the other hand usually introduces a rather pronounced temperature profile for the turbines. The radial component of the profile is intentional in order to thermally unload the hub and root section of the turbine blades.

Because of the pressure and temperature profile problems which have to be simulated for the rig testing it is usually very desirable to have for the rig testing the same instrumentation as installed in the development engines.

4.4 SECONDARY AIRFLOWS

Apart from the primary airflow down the main path through the engine there are a number of secondary airflows which can affect the performance of the components and should therefore if at all possible be included in the assessment and rig testing of the components. Typical secondary flows are:

- Anti-icing air to the front part of the engine
- Air bleed to unload the compressors and to improve handling under critical conditions
- Air off-takes to drive accessories
- Cooling air for the hot parts
- Internal recirculation of air flows due to non-perfect sealing of high pressure areas
- Air for compartment pressurisation

It is in many cases essential to separately declare power losses due to wall friction, ventilation, bearing friction and air pumping in the case of air cooled turbines. Sometimes these parasitic losses are included in the component efficiencies. The inclusion of all the secondary air flows in the engine computer model is a must, their simulation on the various component rigs highly desirable. The largest influence of secondary flows on component performance is usually found with air cooled turbines which can in some cases lose more than 3 % efficiency relative to the ideal aerodynamic rig without any cooling flows.

4.5 VARIABLE GEOMETRY

In some cases variable geometry in terms of variable angle stators in both compressors and turbines is employed. While the variable compressor geometry is usually used to improve the handling or to reduce the noise level in the case of fans, variable nozzle guide vanes on low pressure or free power turbines can be used as active variable geometry in order to influence the cycle data. In either case no extra porblem is being introduced as long as the variables are scheduled as a function of a non-dimensional parameter for instance as $f\ (N/\sqrt{T_t})$. In this case the resulting performance characteristic can still be presented in non-dimensional terms. In the more general case, however, where the variables are being controlled as a function of a dimensional parameter, a number of destinct characteristics for a number of stator angles has to be generated and automatically interpolated in the engine computer model. Only in some special cases where the change of the characteristic is small for the full range of the variables can a procedure be used where the important parameters of the component characteristic are being scaled as a function of the actual guide vane angle. Such scaling factors have usually to be presented as a function of both stator angle and $N/\sqrt{T_t}$ as well.

5. EXTENSION OF THE ENGINE MODEL TO COPE WITH TRANSIENTS

The computation of transient performance is always based on establishing the positive or negative excess torque of the turbine over the compressor at any instance of time. This excess torque together with the total polar moment of inertia of the shaft is then turned into an acceleration or deceleration rate respectively. A simple step by step integration process is used in order to determine the shaft speeds for the next point after Δt [s] has elapsed. The performance computation to be carried out at each point of time is virtually the same as for steady state conditions with the following exceptions:

- The heat exchange between the gas flow and the engine components should be allowed for. It has been found that up to 25 % of the overfuelling during a fast acceleration can be absorbed for the heating up of the blades, casings, liners, rotors etc.

- The different expa   lead to transient clearance changes with a resulting effect on the transient performance of the   nts.
- The heat exchange in the compressors and turbines is similar to the effect of interstage cooling or heating respectively with a resulting effect on the efficiency of up to ± 2%.
- In some cases the high frequency regime is important. Here the interstage volumes have to be taken into account in order to simulate the small delays occuring when filling up volumes with gas due to pressure and temperature changes.

Besides the normal transients between idle speed and full power the power assisted start up process is of particular interest. In former years a clever estimate of the starter motor size required was usually sufficient. Modern engines with their hihgly loaded compressors, however, need in many cases much more attention being given to the start up performance. For the simulation the component characteristics are required right down to very low speeds. Experience has shown that it is in many cases prudent or even essential to test the compressor performance in the low speed regime in a manner which is equivalent to operation of the compressor during start up. Some compressors exhibit double characteristics in this regime depending on how a particular operating point is being approached. A good simulation based on realistic component data does usually give a realistic indication of the surge margin required in this regime. On a number of recent projects compressor development work had to be carried out in order to improve the low speed performance for starting. If an engine is in trouble with respect to starting it usually hangs up at a fairly high speed which cannot be exceeded even if more fuel is being scheduled because the temperature limit is quickly reached or even exceeded. In such a situation only three measures can help. They are:

- More powerful starter motor with high cut out speed
- Compressor bleed either interstage or at delivery in order to unload the compressor
- Compressor development work to improve the low speed characteristic

In any case it is essential to build and instrument the compressor rigs such that at least the core engine compressors can be tested down to very low speeds of say 10 % of design speed.

6. EXTENSION OF THE ENGINE MODEL TO COPE WITH WINDMILLING

The windmilling performance of an engine can be very important for both the altitude light up ceiling and the power off-take capability to satisfy the emergency requirements with respect to the control of the aircraft in the case of an engine flame out. The simulation of the windmilling performance is possible but asks for special information on the compressor and turbine component performance. There are distinct differences in the modes of the component operations when applying a ram ratio across the engine without generating a temperature rise in the combustion chamber. Two classical cases are possible depending on the ram ratio:

Case I: The compressor works in the turbine mode and delivers power to the turbine. Whether the turbine generates any pressure rise depends on its efficiency level under these conditions. This mode of operation therefore represents the opposite of the normal mode of operation with the combustion lit.

Case II: The compressor or fan absorbes some energy from the air stream but does not produce any shaft power. The pressure loss is only so big that there is still some pressure drop across the turbine resulting in some positive shaft power. This power is being absorbed by the compressor whose speed will therefore be somewhat higher than for the case of zero power transfer.

It should be noted that under windmilling conditions the direction of rotation is the same as under normal operation. It can easily be seen that the standard definitive of the isentropic efficiency $\eta_{is\ compr} = H_{is}/H_{eff}$ for the compressor and $\eta_{is\ turb} = H_{eff}/H_{is}$ for the turbine cannot be used any longer for windmilling conditions. It has been found that a suitable presentation of compressor performance when dealing with windmilling can be obtained by replacing P_{t2}/P_{t1}

by H_{is}/T_t and by H_{eff}/T_t. Therefore:

$$H_{is}/T_t = f_1 \, (N/\sqrt{T_t}\,;\ M\sqrt{T_t}/P_t) \qquad \text{and}$$
$$H_{eff}/T_t = f_2 \, (N/\sqrt{T_t}\,;\ M\sqrt{T_t}/P_t)$$

The most suitable presentation of the turbine performance is:

$$M\sqrt{T_t}/P_t = f_1 \, (N/\sqrt{T_t}\,;\ H_{is}/T_t)$$
$$H_{eff}/T_t = f_2 \, (N/\sqrt{T_t}\,;\ H_{is}/T_t)$$

This type of presentation will also lead to managable and symmetrical matrixes of the component characteristic in the computer program.

Apart from the slightly different component performance presentation required, additional information is needed which cannot usually be derived from the running of the components on the normal rigs. Therefore this part of the component characteristic must either be obtained by an intelligent extrapolation or by testing the compressors on a kind of turbine rig and vice versa.

7. RESULTING REQUIREMENTS FOR COMPRESSOR AND TURBINE PERFORMANCE PREDICTIONS

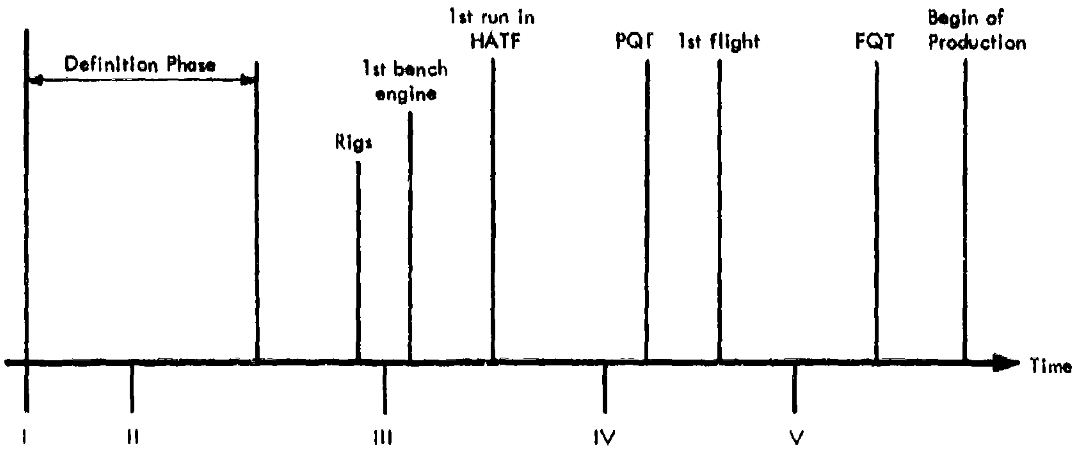
Before summarizing the requirements for compressor and turbine performance predictions it should be made absolutely clear that the component specialist must feel responsible and therefore try to predict the performance of his component when working in the engine. The component rig must only be looked at as a tool to assist in achieving the required component performance in the engine. Experience has shown that in many cases the more advanced highly loaded components become more and more susceptable to the various installation effects. Very great care has also to be taken when an aerodynamic layout is being turned into an actual engine component by the designers. The aerodynamicists must watch and influence very carefully the design compromises that have to be made. Only after the design layout has been settled can the component specialist define the target performance of his component. It is in his interest to make sure that the component rigs are being built as identical as possible to the actual engine component, making allowance for things like exit profiles from the upstream component, struts, swan necks, secondary air flows, realistic clearances to name just a few.

Another very important point to watch is the definition of adequate instrumentation in both the engine and on the rig. The component specialist should have an influence on the type and arrangement of the engine instrumentation in order to allow him to properly analyse any type of aerodynamic problems that might occur on his component in the engine under realistic operating conditions. As far as the original prediction of component performance is concerned the following data is required:

- Full performance coverage for the range between idle and full power
- At least a good estimate of the range between zero speed and idle
- At least a good estimate of the windmilling regime
- Influence factors on clearance changes
- Influence of the Reynolds Number
- Influence of the distortion level
- Information on the growth potential, mainly in terms of mass flow at a speed in case a rematch of the engine is required at a later stage
- Estimate of the noise level generated at the relevant operating conditions

It had been indicated already that this information has to be updated at certain milestones in the program particularly in the light of relevant rig results. It is usually highly appreciated by the performance area if the component characteristics produced are smooth and suitable for trouble free interpolation in the computer.

It is generally accepted that the requirements listed can only be satisfied if the component specialists can base their predictions on some kind of test results obtained from either a research or a demonstrator program with at least similar components. In the U.S. it has been standard practice for many years to only embark on an ambitions engine program after having staisfactorily performed a demonstration program with an engine made up of components close to those of the final product. In Europe the engine manufacturers usually have to demonstrate at least the aerodynamic potential of the assumed component performance with the aid of a component research program before being given the go ahead for a new engine.



I: First guess of component performance based on results from a suitable research or demonstrator program

II: Definition of component target performance

III: Definition of realistic component performance for first engine to test

IV: Definition of component performance as analyzed from engine runs for PQT engine

V: Definition of component performance for production engines

Fig. 1: Approximate Time Schedule for Engine Development Program

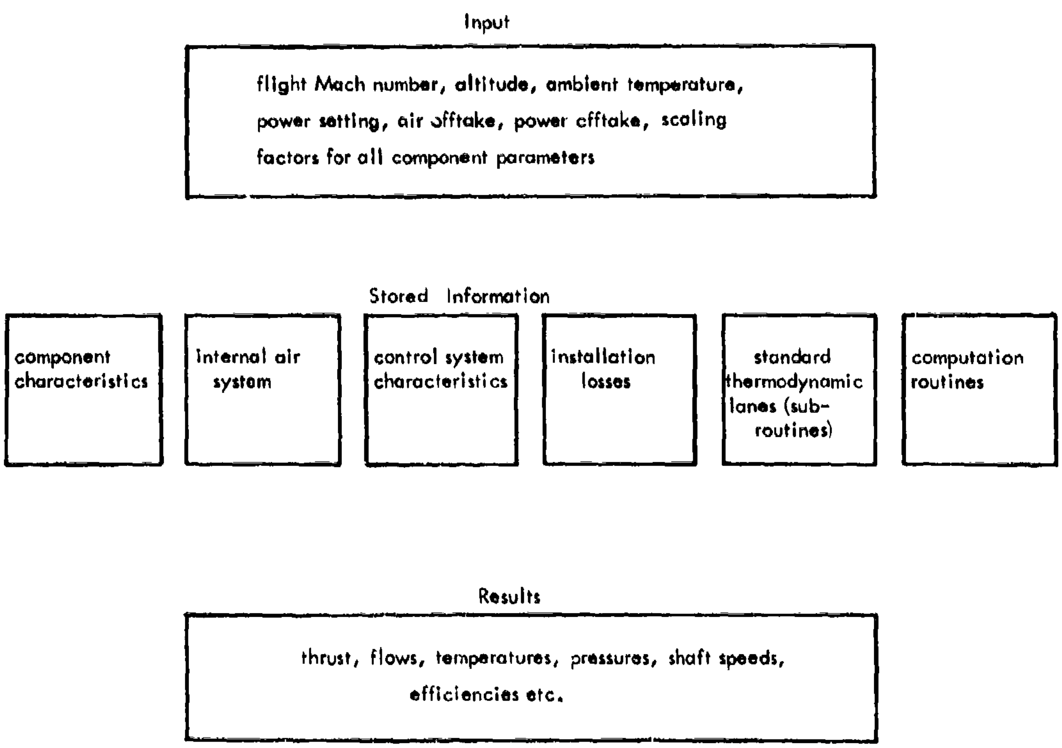


Fig. 2: Structure of Engine Computer Model

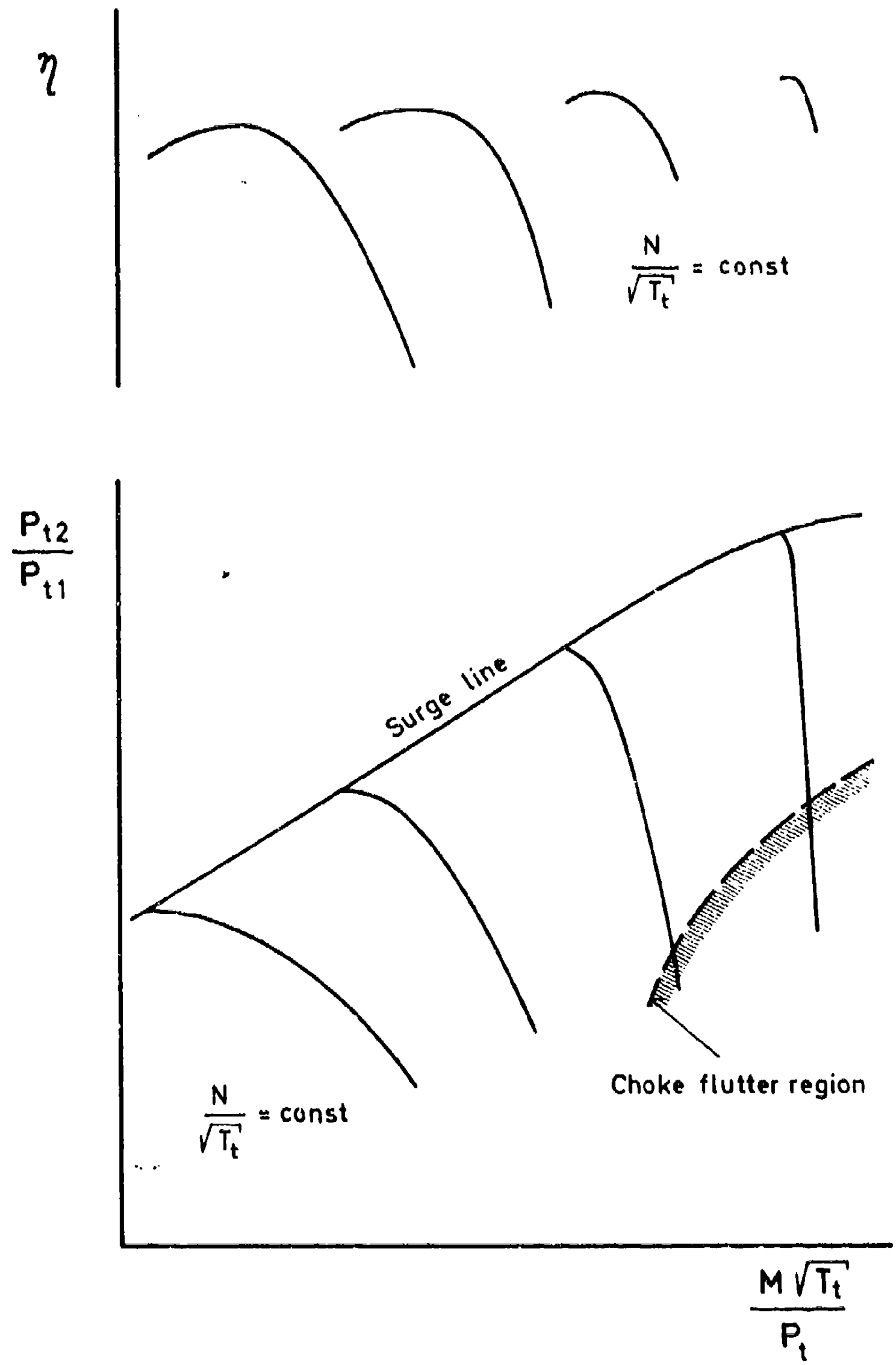


Fig. 3: Compressor Characteristic

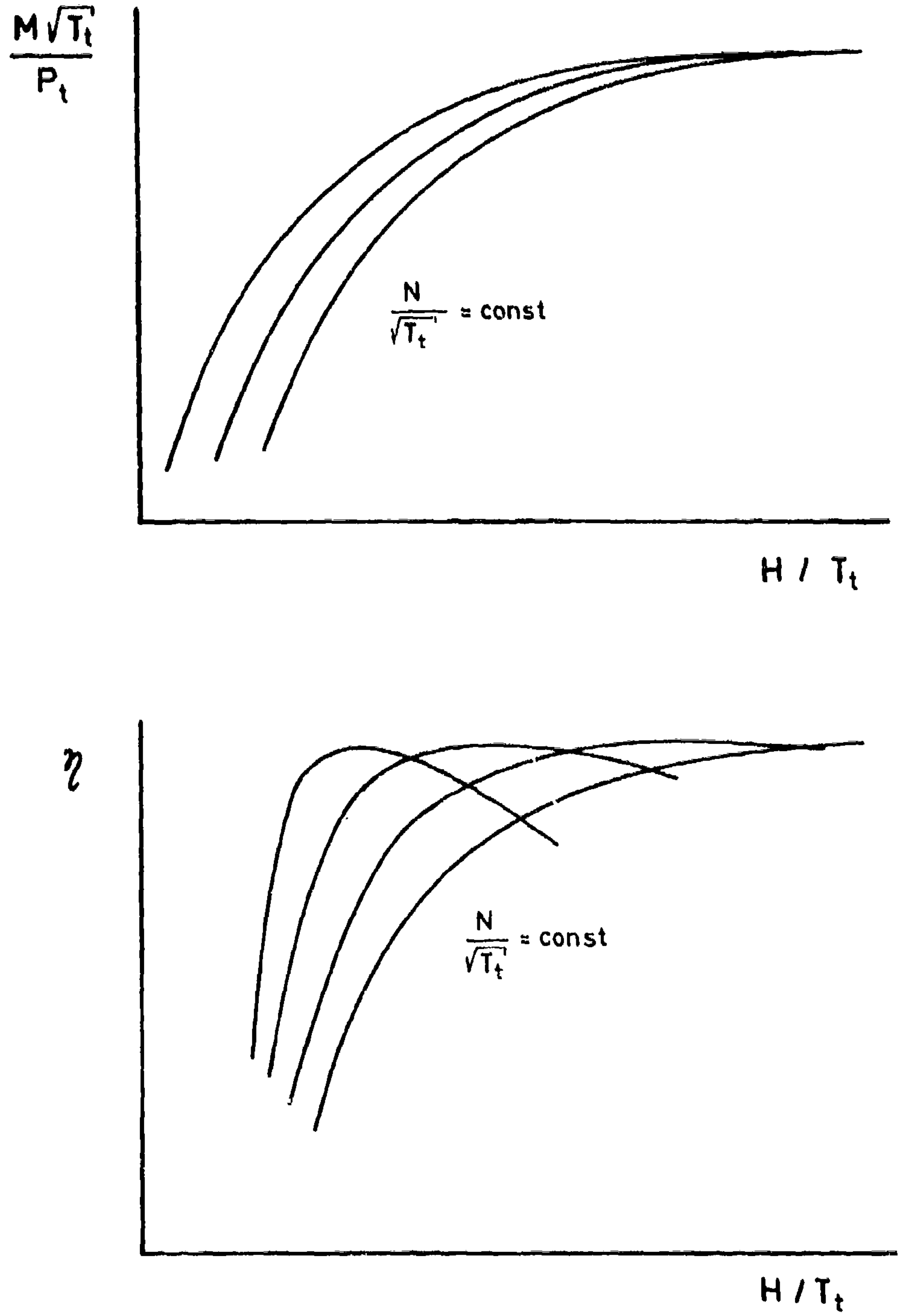


Fig. 4: Turbine Characteristic

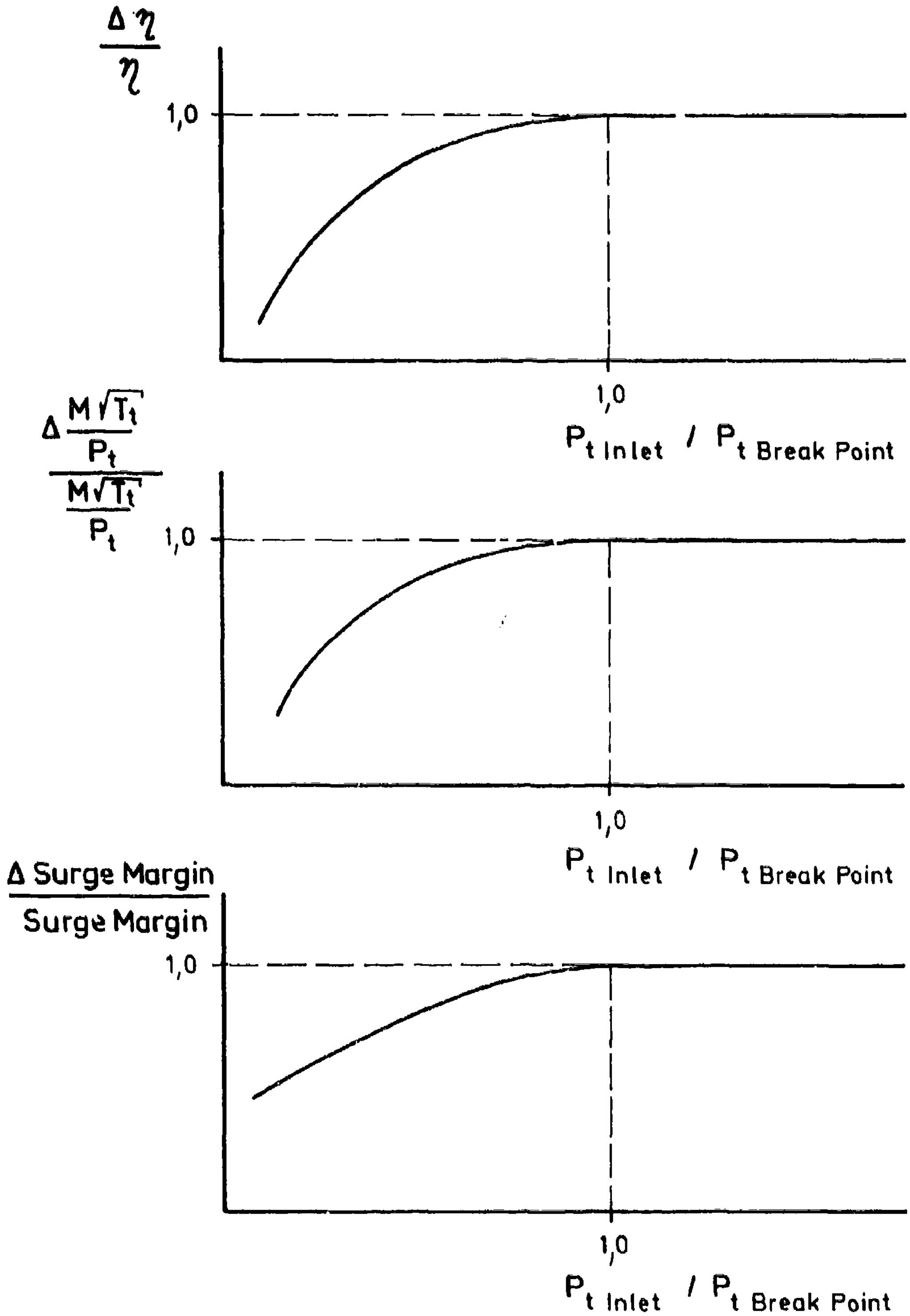


Fig. 5: Approximation of Reynolds Number Effect on Component Performance

COMPRESSOR AND TURBINE PERFORMANCE PREDICTION SYSTEM DEVELOPMENT--LESSONS FROM THIRTY YEARS OF HISTORY

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SUMMARY

This lecture reviews methods developed for prediction of aerodynamic performance of aircraft propulsion system turbomachinery configurations. First, it traces progress in the two classes of methods which can predict only overall performance characteristics or maps. These methods were conceived at least thirty years ago and are not only used, but continue to be the subject of research today. Following this discussion, prediction methods which include flow field definition in the blade passages of compressors and turbines are described. This class of methods, of more recent origin, has been extremely difficult in both an aerodynamic and computational sense. It is, however, the class which will be most rewarding in the long term.

SYMBOLS AND NOTATION

A	flow path cross-sectional area
A_F	frontal area
D_{EQ}	blade section loading parameter (ref. 82)
H	total enthalpy
N	rotational speed
P	total pressure
PR	pressure ratio
r	radial coordinate
r_c	radius of curvature
T	total temperature
U	blade speed
V	velocity
w	flow rate
z	axial coordinate
δ	ratio of total pressure to standard sea-level value
ϵ	function of specific heat ratio (ref. 97) or stream-surface slope (Fig. 8)
η	adiabatic (isentropic) efficiency
θ	ratio of total temperature to standard sea-level value
θ	circumferential coordinate (Fig. 8)
θ_{CR}	squared ratio of critical velocity at turbine inlet to standard sea-level critical velocity (ref. 97)
φ	flow parameter
ϕ	flow coefficient, $\left(\frac{V_{axial}}{U}\right)_m$
ψ	pressure coefficient, $\frac{\Delta H_{isentropic}}{U_m^2}$

Subscripts

in	compressor inlet
m	mean radius

1. INTRODUCTION

In one of the 1945 Institution of Mechanical Engineers lectures on the early development of British aircraft gas turbines, Cheshire (ref. 1) tells us that: "The basic compressor design of the early Whittle engine was fortunately such that the 'matching' problem outlined above did not arise until these engines had been flown. No apparatus existed for running the compressors separately and determining their characteristics, hence, since the operating line only is obtained during the bench running, there was no possibility (apart from a slight and almost valueless variation permitted by changing the propelling nozzle) of knowing where this line was with respect to the compressor characteristics.

"As no surging trouble occurred during the early stages in the development, however, and since (as will be shown later) the operating line approaches the surge line with increasing speed, the compressors generally worked near the optimum efficiency values."

A few paragraphs later in the Cheshire lecture we find other comments like those found in some more current descriptions of engine development programs: "Surging, as first encountered with the W2B type, took the form of a series of loud reports accompanied by violent shaking of the engine, test frame, and test house doors, together with a corresponding oscillation of all instrument readings, when the engine was running at 14,500 r.p.m., the design full speed being 16,500 r.p.m. . . . As it was a great number of diffuser and engine experiments were necessary before the essentials could be established. A series of mechanical failures to impellers and diffusers served both to cause delays and to cloud the issues. . . ."

During the period of more than thirty years between the events described by Cheshire and the present, the issues have frequently been "clouded" in engine development programs. In many cases the problem within the cloud has been associated with unexpected characteristics of the performance of fans, compressors and turbines.

In the first lecture Dr. Bauerfeind has defined broad and exacting current requirements for prediction of compressor and turbine aerodynamic performance. These requirements are partly the result of development of large-scale computer-based methods for simulation of the operating characteristics of air-breathing gas turbine engines. One of the objectives of the Lecture Series should be to make sure that modern performance prediction methods for turbomachinery are clearly understood by users of engine simulation systems, so that capabilities of the prediction methods are fully utilized. At the same time weaknesses in performance prediction should be exposed so that improvements may be planned for the future and so that not too much will be expected at present.

We will also, in this and subsequent lectures, be looking at turbomachinery performance prediction from the point of view of turbomachine design and development specialists. The nature of the methods which are of most interest to them should be known to those concerned with other aspects of propulsion system development.

The objective of this lecture is to give a general view of the progress made in methods for fan, compressor and turbine performance prediction during the period since 1945, and to point out how requirements for prediction have changed during that time to include new aerodynamic and aeromechanical characteristics of performance. The lecture should tell you why the methods to be discussed by Mr. Wall, Mr. Novak and Dr. Japikse are among the best in existence today, and why the development methods presented by Mr. Habrard will continue to be needed in the forseeable future.

2. ELEMENTS AND CONTENT OF COMPRESSOR AND TURBINE PERFORMANCE

2.1 Overall Component Performance

Compressor and turbine performance prediction has two separate but related meanings for the turbomachine specialist. In the context of the first meaning we include estimation of the component overall performance map, as represented by the axial-flow compressor characteristics shown in Figure 1 (ref. 2)*. This performance map describes a number of important features of the equilibrium operation of a particular compressor, including pressure ratios and efficiencies which will be developed by the compressor for a range of flow rates at a given rotational speed. It shows that at each rotational speed, a maximum flow rate seems to exist. The map also shows that at each rotational speed there is a lower-flow limit on stable aerodynamic operation defined by the surge line. It should be remembered that maps of this type describe the performance of a fixed-geometry turbomachine, or at least the performance of a turbomachine with a specific geometry for each individual operating point. For example, one or more blade rows may be set at different angles for low rotational speeds than for higher speeds or bleed ports may be opened at some speeds and closed at others on a given map.

*A parallel description could follow for a centrifugal compressor, or for an axial-flow or radial-inflow turbine.

Maps such as Figure 1 omit numerous significant characteristics of the operating behavior of a turbomachine. As a starting point, Figure 1 does not show effects on performance of changes in gas properties such as molecular weight and specific heat ratio. Neither does it show the effects of Reynolds number and inlet turbulence on performance. By correct utilization of the complete similarity parameters, maps such as Figure 1 can be used to predict the performance of a machine which is in all respects geometrically similar, but only if Reynolds number and inlet turbulence effects are negligible or known. In all cases in which geometric similarity is called for, it must be recalled that this means exact geometric similarity, including similarity of tip clearances, blade section geometry (e.g., roughness, leading-edge thickness) and sealing arrangements. One additional significant element of the overall performance of a compressor is revealed by Figure 1. The general internal operating stability of the compressor will be affected by the presence of regions of unsteady flow caused by flow separation. In compressors, the well-known phenomenon of rotating stall can be an aerodynamic and aeromechanical factor of importance.

Figure 1, however, does not show in any way the effects on performance of modified entrance flow distributions such as distorted or transient inlet pressure, temperature or velocity patterns (Fig. 2). Neither does it show the influence of differences in discharge boundary conditions which ordinarily exist between a test installation and the engine system. Finally, it is impossible on ordinary maps such as Figures 1 and 2 to locate regions in which the various forms of blade flutter may exist and to determine whether (ref. 3) the stresses induced will cause failure (Fig. 3).

The characteristics of turbomachine operation outlined above will be determined at some time during propulsion system design, development, integration, service introduction or operation. If they are not determined by reliable prediction, then problems will be revealed and eliminated only by more dangerous and costly experimentation as the system moves from design toward operation. Sections 3 and 4 of this lecture are devoted to a historical review of the development of our ability to predict the elements of compressor and turbine performance in the design phase of the propulsion system development process.

2.2 Flow Field Definition

The second meaning of performance prediction involves the detailed compressor or turbine internal flow field associated with each operating point on the overall performance map. This aspect of performance prediction can be thought of as an extension of the computation methods used for determination of compressor and turbine design point flow path and blade row geometry.

Flow field definition or design analysis computation methods use as input the turbomachine geometry. The output includes distributions of fluid properties and velocities throughout the compressor or turbine for specified off-design values of flow rate and rotational speed. Advanced programs of this type determine distributions not only at the entrance and exit of each blade row, but also in the blade-to-blade passages within each row. In fact, these blade-to-blade distributions are the necessary input to some systems for estimating flutter characteristics and other aeromechanical features of blade row performance. Internal flow fields have been studied for nonuniform compressor and turbine entrance flows using models which are referenced in Section 5.

Design analysis systems are an important means for optimization of design geometry and for locating requirements for variable geometry. They are, as compared to methods used only in prediction of overall performance maps, time-consuming and capable of generating so much information that the results become difficult to evaluate.

It is evident that by integration of the properties and velocities defined by flow field computations, an overall performance map may be assembled. Turbomachinery aerodynamicists commonly consider flow field definition programs to be the uitlmate in elegance. These programs are also the ultimate in computational difficulty, and it will be part of the purpose of Section 4 to describe past efforts so as to establish a base for Mr. Novak's recent and continuing work.

3. METHODS FOR GENERATION OF COMPONENT PERFORMANCE MAPS

For equilibrium-state and for transient simulation of aircraft gas-turbine propulsion systems, we have observed a progression during thirty years from simple trial-and-error or graphical methods for compressor-turbine matching to very complex digital, hybrid or analog computer simulations.

A look back toward the pioneer reported work on component matching should include descriptions of matching techniques by Kühl (ref. 4), Goldstein et al. (ref. 5), Cohen and Rogers (ref. 6), Morley (ref. 7) and Hodge (ref. 8). Applications to specific cycle studies were covered for the equilibrium-state case by Mallinson and Lewis (ref. 9) and Goldstein et al. (ref. 5), and for the dynamic case by a few reported studies such as Otto and Taylor (ref. 10).

In recent years we find that almost every propulsion research and development activity has developed computer simulation methods for complex engine configurations. Both equilibrium and dynamic simulations are well-documented and tested (refs. 11 to 26). Applications to cycle and control system optimization are both interesting and of obvious assistance in system development.

A striking feature of both the pioneer and current matching-simulation investigations (as well as those of intervening years) is that almost all call for the best possible representation of overall fan, compressor and turbine component characteristics in the form of input of performance maps. The mechanism for incorporation of the map data into the simulation varies, and the map itself may vary, but the overall map ingredient is present. For this reason, we need to examine techniques (other than component construction and experimental evaluation) for generation of compressor and turbine maps.

3.1 Qualities of Performance Map Generalization Methods

We must remember that up to about 1945 very few compressor and turbine configurations suitable for propulsion system application had been built, and even fewer had been tested as separate components. The multistage axial-flow compressor was regarded with suspicion by some very influential people. In view of this, it is surprising to learn that there was a reasonable understanding of many of the features of off-design performance for both compressors and turbines. We can find references in the literature to impeller-diffuser matching in centrifugal compressors, stage-interaction effects in axial-flow compressors, and even to the effects of inlet distortion. For example in Kühl's series of lectures (ref. 4) on gas turbine controls he states: "The similarity must extend also to the supply and disposal of the air, that is, the distribution of velocity, pressure, and temperature must be similar especially at the inlet (in general, velocity, pressure and temperature being constant)."

However, up to 1945, the effects of flow path and blade row geometry on performance were certainly not well defined. Therefore it is easy to understand why methods of performance prediction for compressors and turbines were often based on the principles of similitude and on attempts to develop parameters for generalization of the few available performance maps. Good descriptions of generalization methods and of their use may be found in Howell and Bonham (ref. 27), Robbins and Dugan (ref. 28), Kholshchevnikov (ref. 29), and Mellor and Root (refs. 30 and 31) for compressors and by Horlock for turbines (ref. 32). In each of these methods, a reference operating point is utilized and map parameters are generalized with respect to this point. Users of each method are warned that particular generalized plots are associated with families of related compressors, each based on a specific design philosophy. This is of course recognition that no scaling method is of much value when significant geometric distortion is involved.

Nevertheless, the generalization methods of References 27 through 32 are useful for preliminary work. It should again be mentioned that additional empirical corrections to predicted maps would be required for Reynolds number effects, nonuniform entrance conditions and other map-influencing parameters mentioned in Section 2.1 (refs. 33 to 40). There is no way in which generalized map prediction can ' used to estimate rotating stall patterns in compressors or aeromechanical behavior for ge ·ically nonsimilar turbomachines.

Most of the modern engine ·ulation methods use quite simplified forms of compressor and turbine map scaling. An example is found in the NASA digital simulation program DYNGEN (ref. 14), which requires input of fan, compressor and turbine maps by blocks of data giving numerical values of parameters for individual operating points. The program then scales these real maps to the simulated system map by the equation

$$PR = \frac{PR_{design} - 1}{PR_{map,\ design} - 1}\,[PR_{map} - 1] + 1$$

$$w = \frac{w_{design}}{w_{map,\ design}}\,[w_{map}]$$

$$\eta = \frac{\eta_{design}}{\eta_{map,\ design}}\,[\eta_{map}]$$

for the fan and compressor maps (and by similar equations for turbine map variables).

The DYNGEN program has no mechanism for correcting map prediction for Reynolds number effects although other dynamic simulation programs do (for example, ref. 17). It is apparent that if predicted performance maps are used as input to typical simulation programs, they must be remarkably similar to the actual maps of the system under study in order to obtain reliable system simulation results. Dependable performance map generalization methods could serve a more

useful function in these cases if supplemented by empirical corrections for Reynolds number, inlet distortion and other aerodynamic parameters.

There has not been any significant reported progress in compressor and turbine performance map generalization methods during recent years. However, for both centrifugal and axial-flow compressors, the simple and straightforward nature of this approach makes it attractive for propulsion system development applications. Similarly in the case of turbine configurations, performance scaling methods based on the principles of similitude are attractive.

3.2 Qualities of Stage Performance "Stacking" Methods

For compressors, and especially multistage axial-flow compressors, there has been a good deal of attention devoted to prediction of overall performance maps by "stacking" of predicted performance of individual stages. Stacking techniques also have been used quite successfully in compressor performance improvement by rematching. The stacking concept has appeared frequently in the literature throughout the last thirty years and its advantages and problems should, therefore, be considered.

3.2.1 Origin and development of the stacking concept

It is difficult to identify the first utilization of the techniques of multistage compressor performance estimation by stacking. The method is referred to by Howell (refs. 41 and 42) and by Eckert and Schnell (ref. 43) as a method used in the early period of gas turbine development. In 1952, Finger and Dugan (ref. 44) published a detailed description of the steps required to combine estimated or known stage performance curves into an overall map. The stage curves were plotted as stage total pressure ratio and adiabatic efficiency as a function of a stage flow parameter. The total pressure ratio and efficiency were based on mass-average measured values from a single stage test and the flow parameter was determined from the same stage data using the equation

$$\varphi = \frac{w\sqrt{\theta_{in}}\,/\delta_{in}}{U/\sqrt{\theta_{in}}}\left(\frac{1}{A_{in}}\right) .$$

The principal result of the Finger and Dugan analysis was a clear demonstration of qualitative changes in operating points on the individual stage curves corresponding to changes in overall map flow rate and rotational speed for multistage compressors.

Following the Finger and Dugan work, Benser (refs. 45, 46), Huppert and Benser (ref. 47), Stone (ref. 48), and Doyle and Dixon (ref. 49) studied in detail the relationship between stage curves and overall performance with emphasis on stage stall and compressor surge line prediction. Benser utilized stage parameters of pressure coefficient, ψ , and adiabatic efficiency η, plotted against flow coefficient, ϕ. These parameters are defined in the Section SYMBOLS AND NOTATION. Benser also introduced the idea of a noncontinuous and double-valued form of stage characteristic (Fig. 4) as a possible means for good definition of compressor behavior near the surge limit.

Possibly the greatest contribution made by stage-stacking studies during the early years of the 1950-1960 decade was in its use as an aid to modification and improvement of the performance maps of several multistage compressors (refs. 50 to 53). Figure 5 (ref. 51) shows the performance curves estimated by a stage-stacking method and measured for an NACA research compressor. The compressor flow path and blading were modified on the basis of an analysis of individual stage performance and the measured map of Figure 5 represents a considerable improvement over the performance of the original compressor. However, the quantitative results achieved by estimation of performance maps using either generalization or stacking systems must always be applied with care. Figure 6 (ref. 54) shows that this is true even when the most competent organizations are involved. The discrepancies of Figure 6 are typical of those experienced by many organizations up to the present time.

3.2.2 Sources of stage performance curves

Both computed and measured stage performance curves have been used in stage-stacking methods. Computed mean-radius analysis of stage flows as suggested by Howell and Bonham (refs. 27, 41, 42) has been the most frequently discussed means for estimating untested stage characteristics, but some analyses have used integrated stage performance from flow field definition calculations of the type considered in Section 4.

When extensive collections of measured stage performance are available, it is convenient to use performance curves fitted to test data from stages having similar geometric properties. This was the approach followed in most of the initial stacking investigations (for example refs. 44 to 47). It was also used in the construction of stage performance curves for compressor design modification programs (refs. 50, 52 and 53) and the practice has continued through quite current compressor prediction studies (40, 55, 56).

3.2.3 Critique of stage-stacking

The difficulty in estimating stage performance for an untested compressor configuration has presented the greatest problem in obtaining reliable results. Best correspondence between predicted and measured overall maps has been achieved when curves are available for stages which are very similar in a geometric sense and when these curves have been obtained from experiments in a multistage compressor environment.

The reasons for the preceding statement should be apparent. Stage-stacking is nothing more than a classical application of the principles of similitude. If similarity conditions are satisfied, the method works. If the conditions are distorted, the method loses some of its value. The stage performance parameters ψ and ϕ, as applied in stacking lead to a result which only partly accounts for effects of compressibility. The stage-to-stage effects are partially included, but the effects of compressibility on stage flow coefficient range (choking to stall) and efficiency are not properly dealt with. Some attempts have been made to develop more effective parameters for stage characteristics, but these have not been very successful (refs. 57 and 58).

3.3 Recent Extensions of Overall Map Methods in Multistage Compressor Performance Simulation

Both map generalization and stage-stacking methods have been extended and supplemented to match new engine cycle requirements and to improve the compressor and turbine component simulation in the process of cycle and control simulation investigations.

Methods for stage-matching and surge line behavior for variable-geometry cases have been briefly described by Gray (ref. 59) and outlined in some detail by Southwick (ref. 60). The control situation for variable-geometry cases is discussed by Tervo and Tringali (ref. 26).

Most of the more recent engine simulation techniques include some mechanism for accounting for dynamic effects on compressor and turbine operation and behavior. These effects include the influence of heat transfer and storage phenomena, the effects of "swept" volumes (the mechanics and thermodynamics of stage volumes, including axial spaces and cavity flows), and the effects of changes in geometry (for example, tip clearance) with changing engine operating point. These topics are important but will be covered here only by reference to a range of recent reported work (refs. 15, 17, 18, 21, 35, 56, 61 to 63). For its historical value we should mention the 1959 forecast of problems to come by Gabriel and Wallner (ref. 64).

3.4 One-dimensional or "Mean-radius" Performance Prediction for Turbines and Centrifugal Compressors

Before we turn to the subject of flow field definition by computation, it is appropriate to consider the one-dimensional analyses that have been developed and used extensively for axial- and radial-flow turbines and for centrifugal compressors. These configuration types have not been deliberately neglected in this lecture, but the fact is that the vast majority of the published work on performance prediction has been on problems related to the axial-flow compressor. The results realized from one-dimensional approaches have been used with considerable success for turbines and centrifugal compressors (keep in mind that the number of stages involved is commonly one or two).

For axial-flow turbines, the textbook example of one-dimensional performance prediction is the method of Ainley and Mathieson (ref. 65), with improvements and supporting information (refs. 66 to 68). This method is easy to use and must hold a record for citation in the literature. Similar techniques have been described for radial-inflow turbines (refs. 69 to 72) and for centrifugal compressors (refs. 73 and 74). Results can be compared with typical performance map characteristics (for example, ref. 75 and 76).

4. COMPUTATION OF TURBOMACHINE FLOW FIELDS

Section 2.2 introduced the subject of turbomachine performance prediction by computation of the internal flow field at individual operating points described by given values of equivalent inlet mass flow rate and equivalent rotational speed. This is the classical direct problem in which the flow path and blade row geometry are known. However, solutions for the general performance prediction problem should not be confused with similar solutions for <u>only</u> the design flow rate and rotational speed. Design point direct solutions have been reasonably common in the past. We will restrict our discussion to the less frequent attempts to predict off-design flow fields and to integrate these results into performance maps.

The estimation of stage characteristics by one-dimensional (mean radius) calculations was briefly mentioned in connection with overall performance map prediction. Although these methods have often been misunderstood as flow field methods, they are not.

The only realistic models of turbomachine internal flow fields have been those which are described as quasi-three-dimensional or three-dimensional. Until quite recently, nearly all of these models have assumed steady flow relative to the individual blade rows. Additional restrictive assumptions have necessarily been used in order to make the computational problem reasonable. In this lecture we will not reference or discuss a vast range of analyses based on the assumption of inviscid flow. These analyses will only be mentioned in cases where inviscid solutions have made, or can obviously make a genuine contribution to compressor or turbine performance prediction.

Figure 7 (ref. 77) introduces some of the ideas which are essential to the flow field computation problem. This figure is a meridional plane section of a turbojet engine with an axial-flow compressor and turbine. The figure cannot help but remind us of the complexity of the flow under consideration. Figure 8 shows a simplified meridional section of an axial-flow compressor flow path with some of the conventional notation used in flow field methods. Computation of the meridional plane flow field or of the flow in a hub-to-tip surface between blades of a given row has been an element of all quasi-three-dimensional methods. The second element is the computation or estimation of the flow through individual blade rows on blade-to-blade surfaces.

Realistic attacks on the turbomachine flow field computation problem are difficult to identify before about 1955. A paper by Cohen and White (ref. 78) has been mentioned in the literature, and it appears to include discussion of some of the important computational difficulties encountered in flow field definition. The same is true of the 1949 paper by Merchant (ref. 79). Both of these papers as well as those reporting independent efforts by Serovy (refs. 80 and 81) and Swan (refs. 82 and 83) were concerned with axial-flow compressors. For historical purposes Figure 9 shows the character of the stage curves produced by Serovy and Swan. Both Serovy and Swan used digital computers to achieve solutions and both used a steady, axisymmetric model of the flow (a quasi-three-dimensional approach). Meridional plane computations were made by Serovy using a radial equilibrium condition with no allowance for streamline curvature or slope (see Fig. 8). Swan included these terms. Serovy and Swan both used experimental data (cascades and blade rows) to estimate the blade-to-blade plane flow deflections and circumferentially-averaged total pressure losses. Swan used a transonic stage as a test case and therefore included a shock loss computation. The important feature of both these studies was that they were based on iterative solutions of the radial equilibrium and continuity conditions with accounting for accumulation of losses through individual blade rows. The rotor and stage performance curves computed were acceptable. The radial variations of velocity and properties computed at calculation planes between blade rows were not satisfactory. This aspect of the computational problem (the radial distributions) has continued to be important in all subsequent work.

For axial-flow compressors flow-field computations and comparisons with test data have been made by numerous investigators (refs. 84 to 93). R. A. Novak will, in his lecture, describe the results of his long-term and valuable contributions in this area, including those of the men who have worked with him through the years. The basic formulations of the governing equations published by Wu (refs. 94 and 95) and the 1967 paper by Novak (ref. 96) have been the starting point for many of the significant flow field methods.

For axial-flow turbines only a few flow field studies have been fully reported. Flagg (ref. 97) developed a computer-based program and his test cases included a two-stage turbine (Fig. 10). Smith, Barnes and Frost working at the NGTE have also computed turbine and centrifugal compressor flow field cases with success (refs. 98 to 100). Herzog (ref. 101) and Renaudin and Somm (ref. 102) have published flow field results for steam turbines.

5. FAN, COMPRESSOR AND TURBINE PERFORMANCE PREDICTION AS AN UNRESOLVED PROBLEM

There is no part of the performance prediction problem which can now be considered as a resolved problem. This lecture has been historical in nature and the later presentations represent the current status. Place these following lectures in the center of a technological stage surrounded by simulation specialists working with computer models of overall compressor performance; by Novak and Hearsey (ref. 103), Korn (ref. 104), Mazzawy (ref. 105), Mokelke (ref. 106) and others investigating flow field for turbomachines with distorted entrance flows; and by Erdos et al. (ref. 107), Stuart and Hetherington (ref. 108), Anderson (ref. 109) and Briley and McDonald (ref. 110) seriously developing three-dimensional flow field computation models. The situation, like turbomachine flow fields, is difficult to analyze and no one can predict the result.

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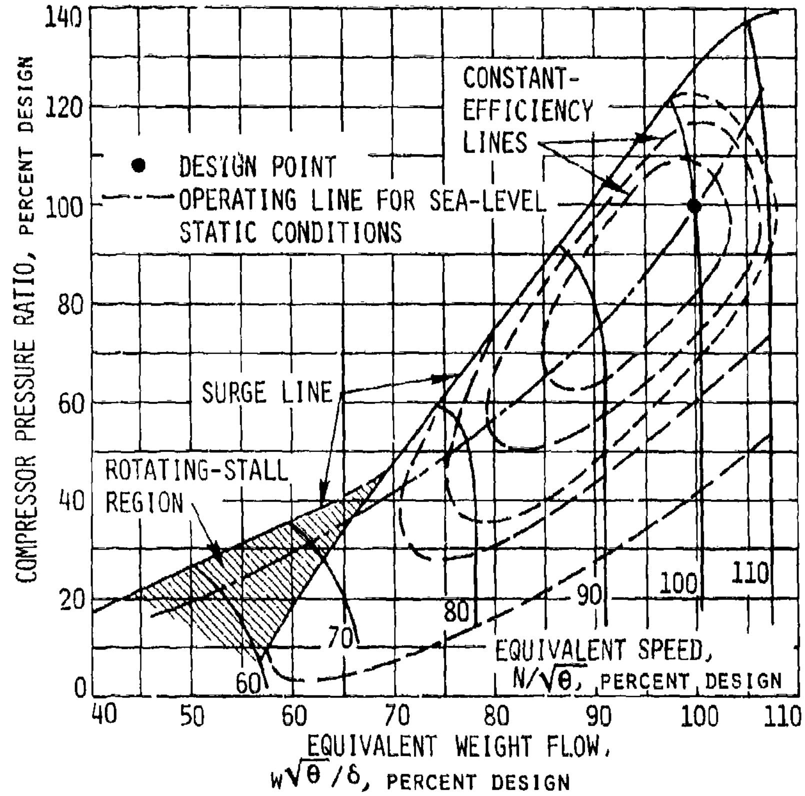


Fig.1 Example of axial-flow compressor performance map (Ref.2)

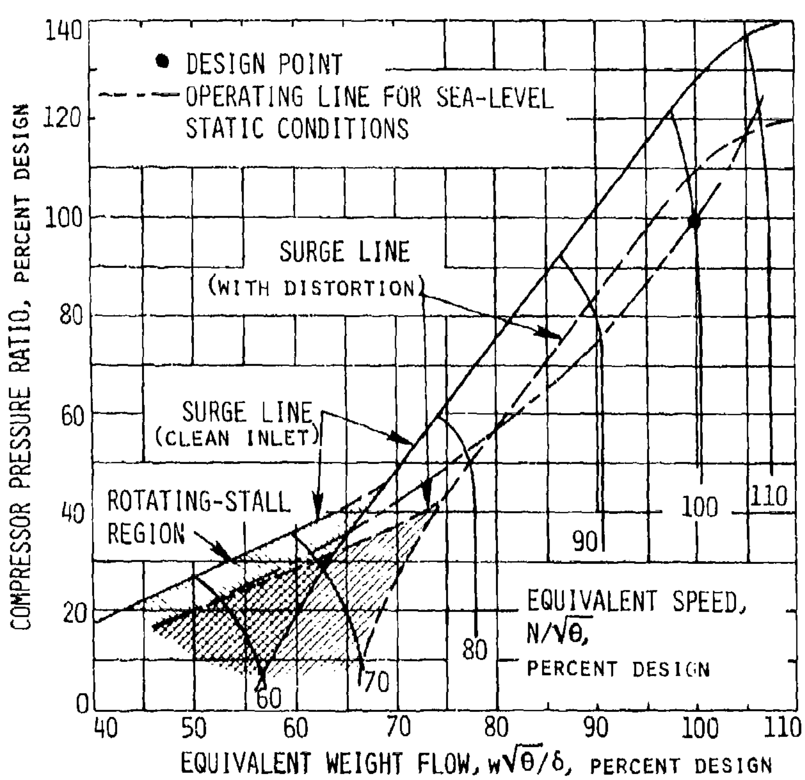


Fig.2 Example of axial-flow compressor map showing effect of inlet flow distortion (Ref.2)

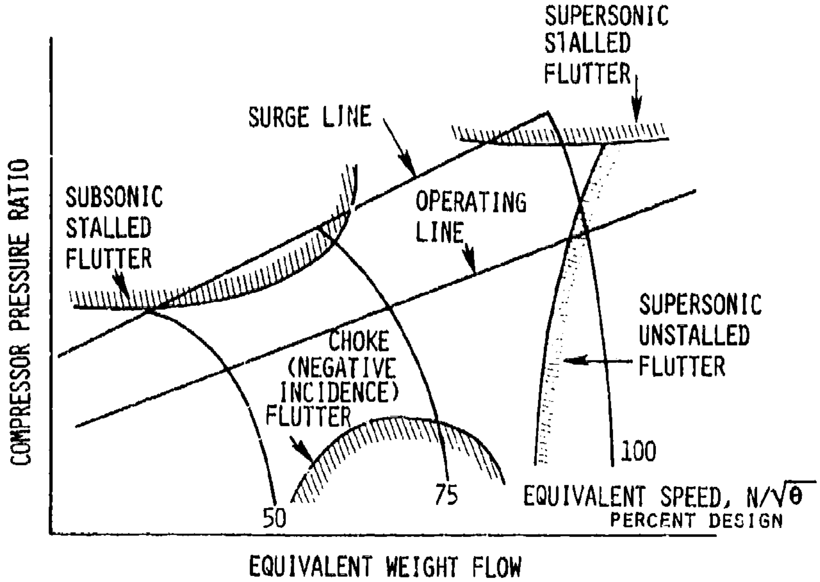


Fig.3 Schematic compressor map showing boundaries for four types of blade flutter (Ref.3)

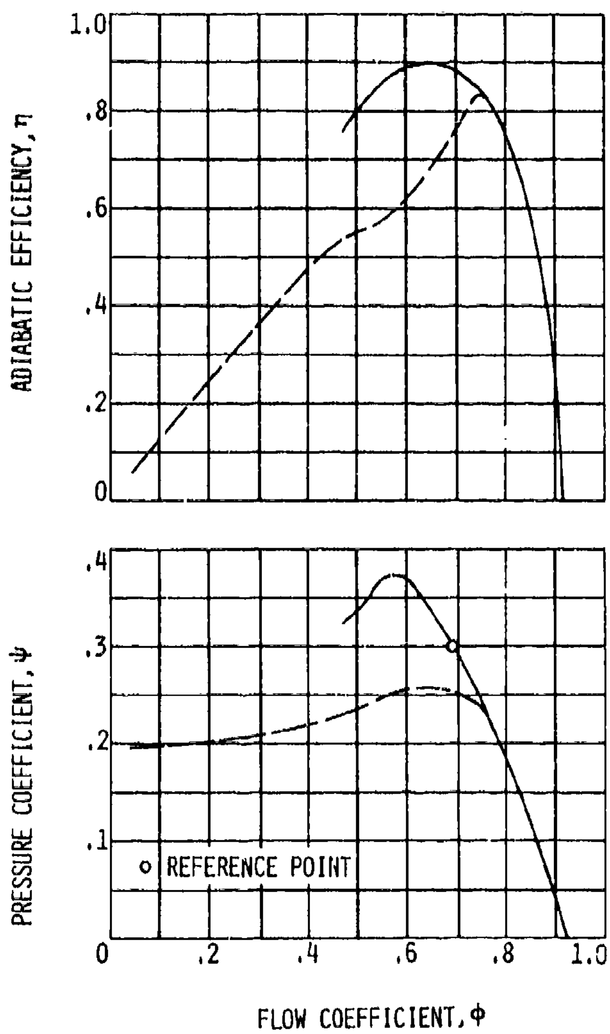


Fig.4 Example of assumed stage performance characteristics used in stage-stacking analysis of Benser (Ref.45)

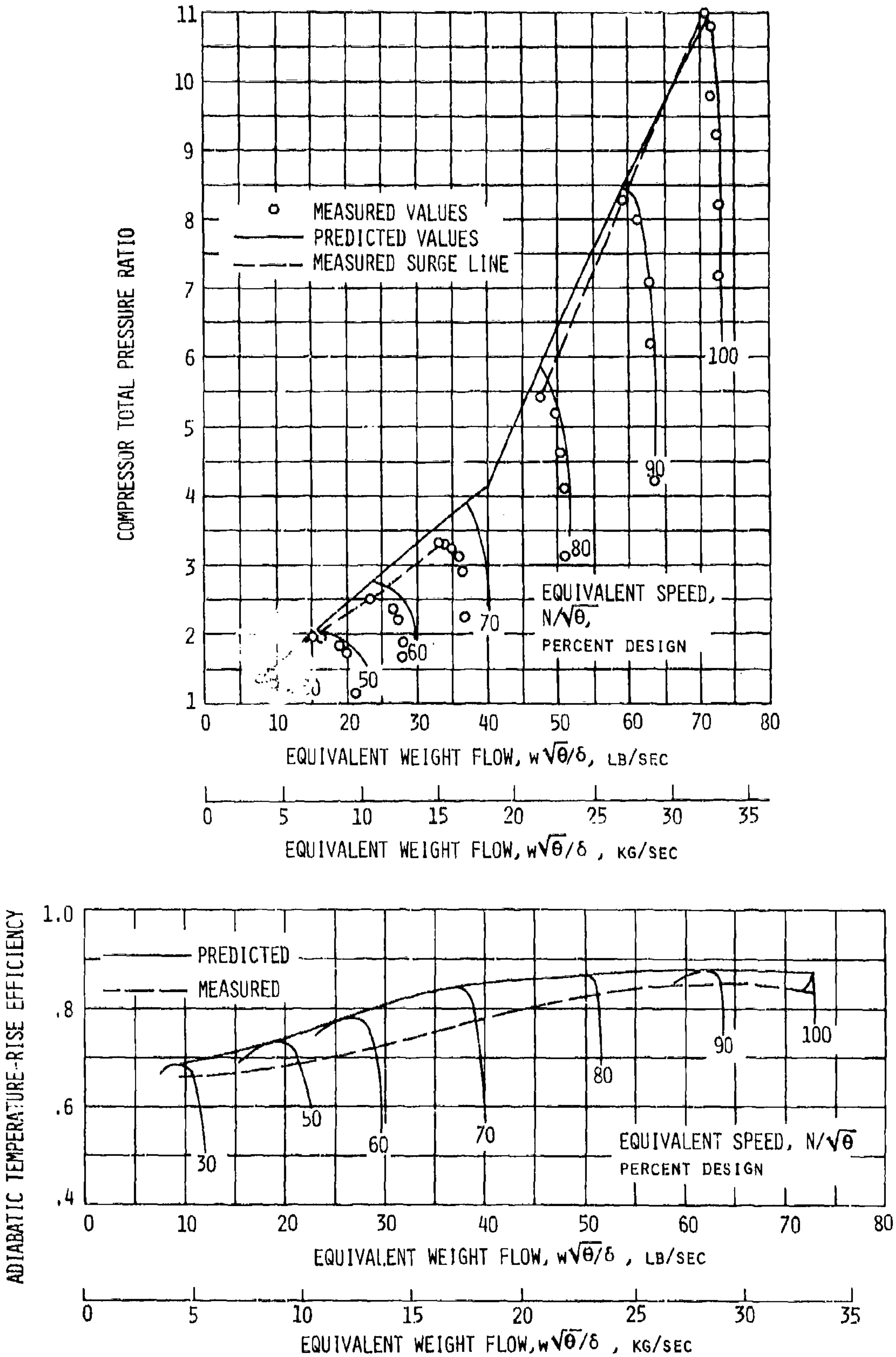


Fig.5 Predicted and measured overall performance for modified eight-stage axial-flow compressor (Ref.51)

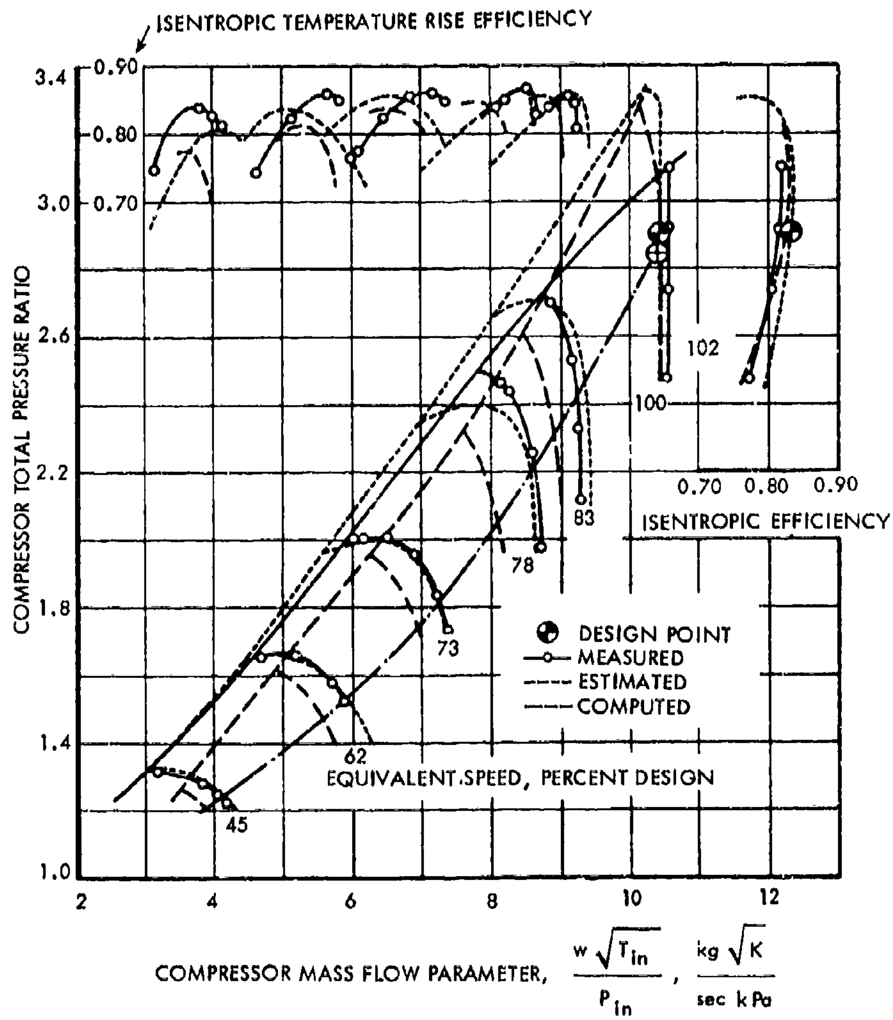


Fig.6 Predicted and measured performance characteristics of the MTU three-stage axial-flow compressor (Ref.54)

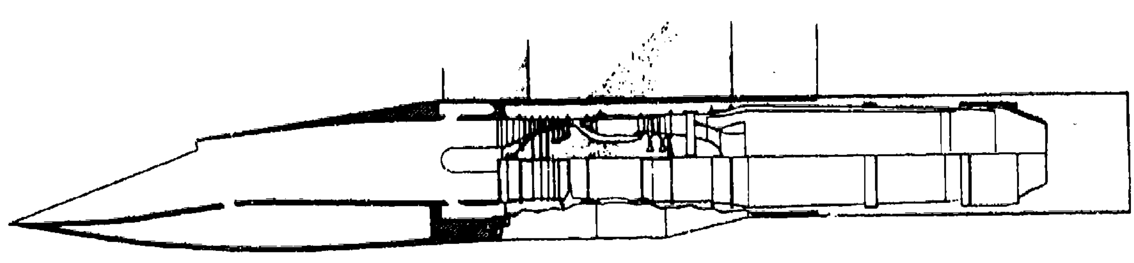


Fig.7 Cutaway of turbojet engine installation (Ref.77) showing meridional plane flow path

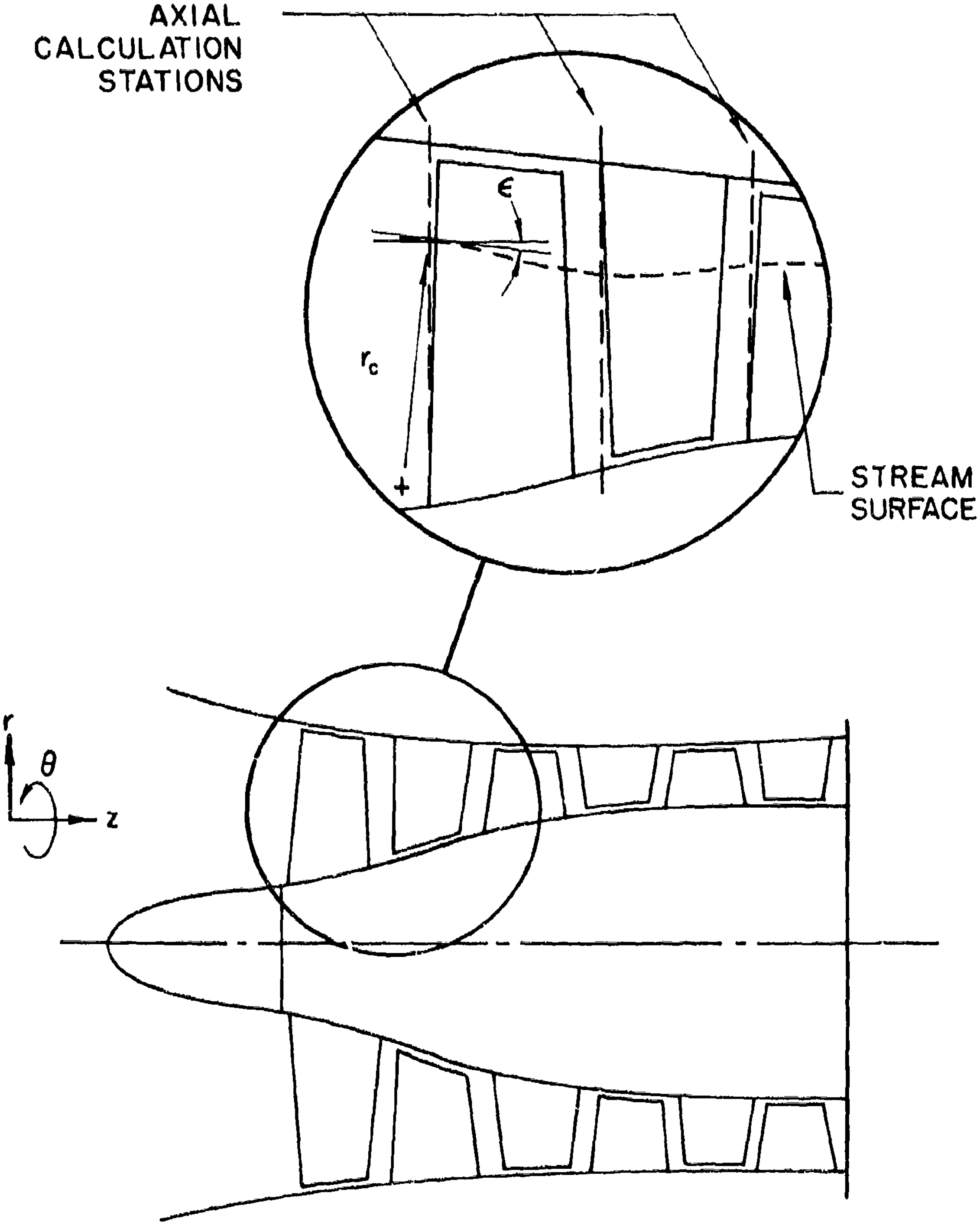
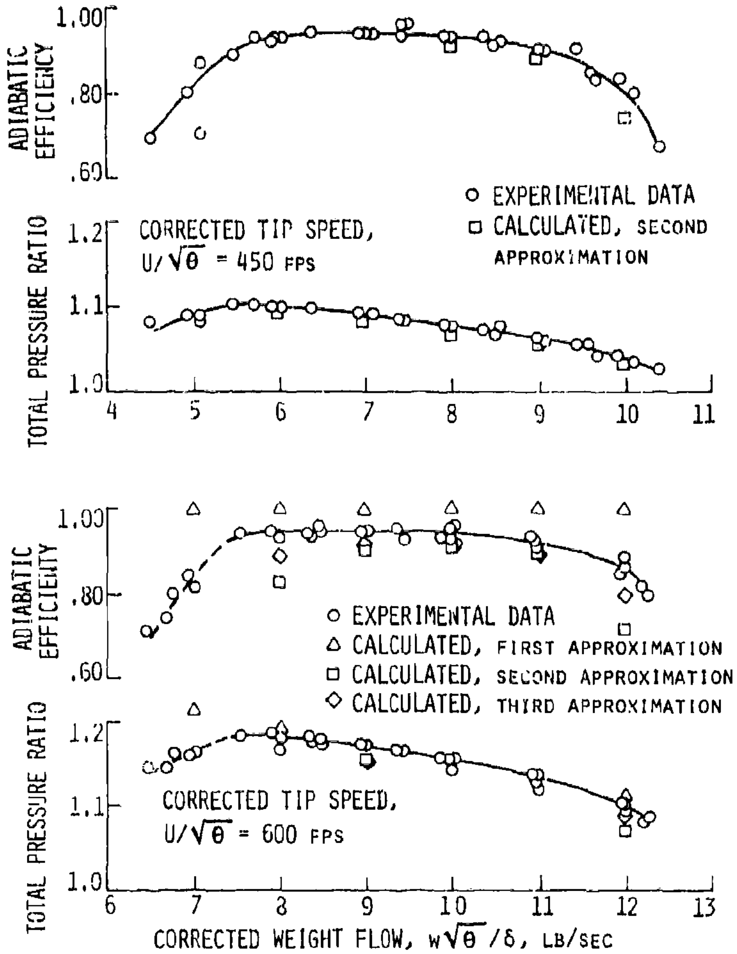
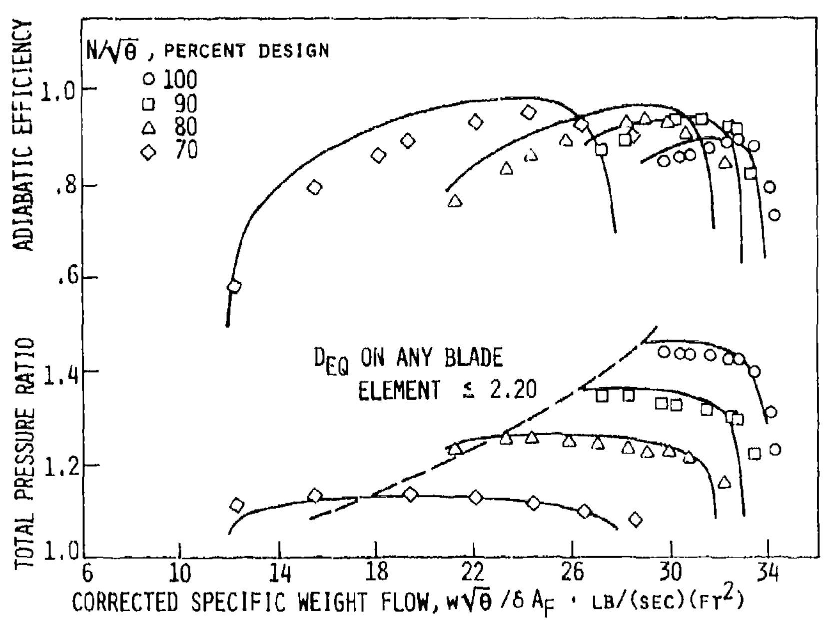


Fig.8 Meridional plane section of axial-flow compressor showing axisymmetric stream surface notation





(B) SWAN, 1958 (REF. 82)

Fig.9 Comparison of computed and experimental mass-averaged stage pressure ratio and adiabatic efficiency values at various flow rates for two flow field prediction systems

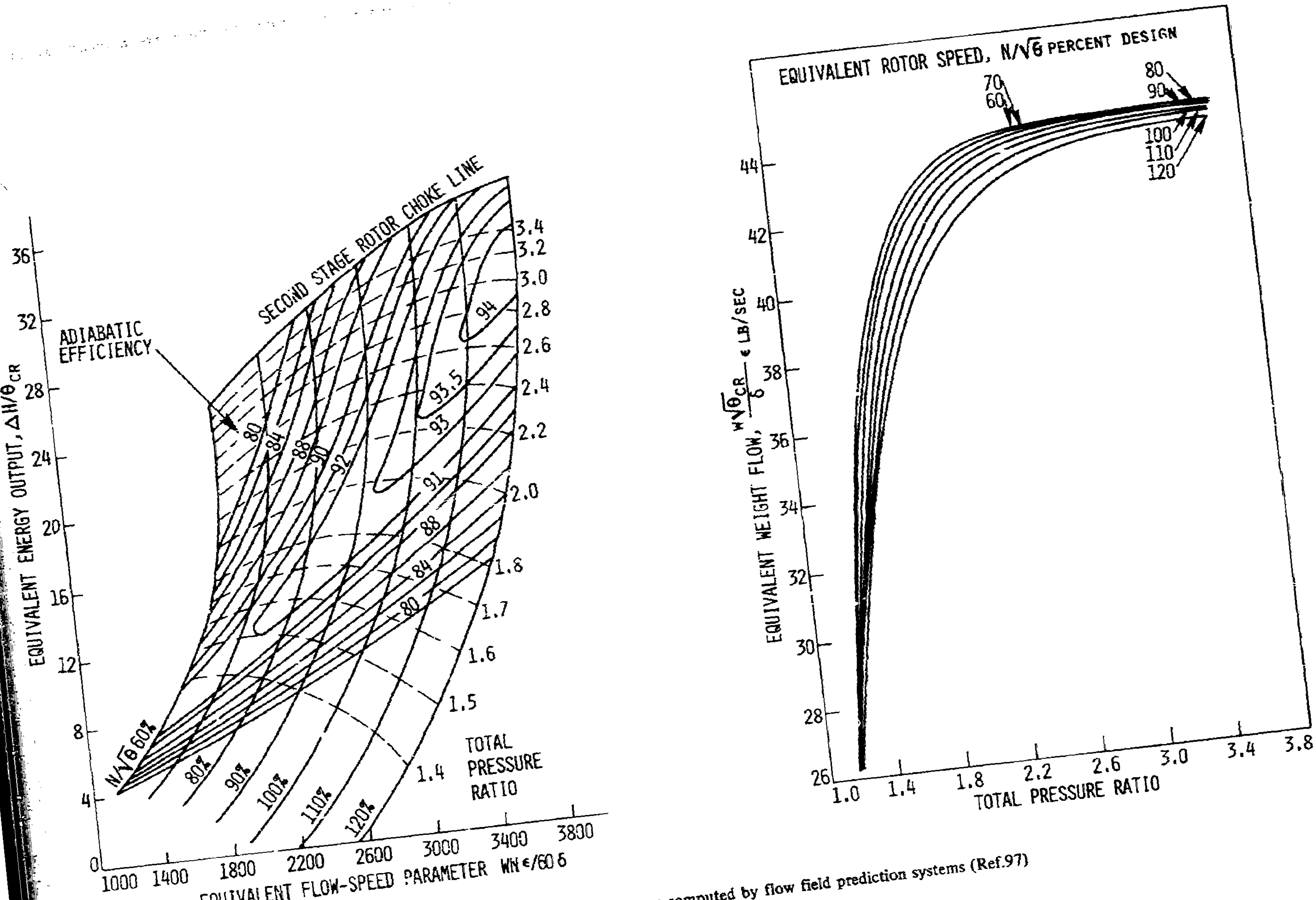


Fig.10 Performance of two-stage turbine as computed by flow field prediction systems (Ref.97)

Axial Flow Compressor Performance Prediction

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Summary

Owing to the nature of axial flow compressors, performance prediction is characteristically difficult to achieve by theoretical analysis, and therefore recourse to gross empiricism, laced with theory, is fundamental to methods employed.

Factors determining the performance requirements of compressors, and the physical mechanisms which control their ability to satisfy these requirements, are therefore discussed to expose the nature of compressors.

Compressor design optimisation is described to illustrate how effective compromise can be achieved between design point performance and various off-design excursions demanded by Turbomachine performance.

Typical performance prediction methods which reflect the nature of compressors and quantify their performance characteristics, are described with some reference to the influence of engine environmental factors.

Potential developments are discussed which could influence the type of design employed in future engines.

1.0 DETERMINANTS OF DESIGN

1.1 Engine Performance Requirements Current applications of the axial flow compressor demand a wide variety of performance capability to suit the particular environment in which they operate. Broadly these include Industrial power applications, high performance military aircraft and long range thermally efficient civil air transport applications - Fig 1.

Industrial applications include the generation of electricity, marine propulsion, gas pumping and locomotive power. Operational reliability with high thermal efficiency is demanded over a wide range of operating conditions. Since industrial applications usually involve the generation of large quantities of energy, the axial flow compressor supercedes the centrifugal compressor owing to its inherent capacity to pass higher mass flows. Large scale centrifugal compressors are enormously cumbersome and difficult to manufacture. Large pressure ratios, up to 40:1, can be generated in machines with outside diameters ranging up to 10 ft. High thermal efficiency can be obtained by using large nos. of stages with correspondingly low stage pressure ratios, since weight is not a consideration in these circumstances. However, in order that such a machine can respond to widely ranging power demands, the use of sophisticated off - design variables is demanded in the form of variable geometry stators and multiple bleed systems. This implies fairly sophisticated off - design variable scheduling which can respond automatically to the demands placed upon it. Much development time must be spent optimising such a complex device with the aid of 'stage stacking' type analysis.

Military applications demand simple lightweight, aerodynamically robust compressors which can respond to military manoeuvre requirements over a wide range of operating conditions. Low level - high M_N flight capability must be combined with high altitude cruise requirements. Comparatively low core pressure ratios (15:1) can only be obtained due to the physical limitations of the engine carcase at sea level. Low bypass ratio configurations are demanded since most of the thrust is obtained by core jet velocity for high M_N flight. However, in order to maintain a thrust boost capability by the use of reheat or afterburner devices, a small bypass ratio (μ = 0.6 - 1.0) is required. This combines the facility for thrust boost at low altitude and high M_N cruise at high altitude. Thus a multistage split flow fan is used in conjunction with a short light weight multistage core compressor. High thermal efficiency is of secondary importance to high performance and light weight.

Civil air transport applications demand high bypass ratio engines for high propulsion efficiency with jet velocity compatible with high subsonic M_N (0.85) cruise conditions at altitude (30,000 ft). High core compression ratios, limited by Turbine Entry Temperatures, are required in order to obtain high thermal efficiency which minimise operating costs. Weight, thermal and propulsive efficiency are all of paramount importance at the cruise condition even at the expense of efficiency at Take Off conditions. Potential operating efficiency (S.F.C) is strongly determined by the bypass ratio chosen and the fan pressure ratio consistent with that - see fig 2. However, a limit to the attainable bypass ratio is reached depending on engine size. Engine core components cannot be realistically scaled down indefinitely without sacrifice of aerodynamic and mechanical efficiency. Shaft bearings, combustor fuel nozzles and turbine blade cooling passages become impractical below a certain size. Thus very large engines (> 60,000 lb thrust) are implied by the use of energy saving high bypass ratio engines, see Fig 3. However this type of configuration is likely to be advantageous from the point of view of noise reduction - low fan tip speeds and jet velocities fundamentally reduce noise output. Note that the trend towards high bypass ratio for human flight is following the pattern of nature. The bypass ratio of the average bird having approx 1 ft dia fan and $\frac{1}{4}$ inch diameter-core is approx 1000:1 with negligible fan pressure ratio, assuming the fan to impart energy to half the frontal area.

1.2 Mechanical Design Constraints Principle constraints which determine the overall compression system configuration in a typical civil type engine, Fig 4, relate to the fan hub inlet and core exit positions. For a given inlet fan area constant with a given thrust level increased fan hub/tip ratio controls the fan inner diameter and therefore disc diameter capable of supporting the fan with a long fatigue life. Larger inner dias. increases the space between blades and therefore the disc rim area to support it. For a given fan tip speed the fan root, speed is therefore determined and therefore the amount of fan root pressure ratio consistent with aero loading limitations. At core exit, the physical environment (temperature $\sim 800^{\circ}$K) controls the material properties and therefore max. stress levels and fatigue life. Thus a max. compressor rim speed is found which determines the non dimensional work output per stage in the core compressor (1100-1200 ft/sec) depending on the mechanical design). Additionally, a hub/tip ratio limitation is generally imposed ($\not> 0.95$) for a given flow area, so that blade end losses do not become a significant determinant of component efficiency. Thus it remains to fill in the space between these end points in a manner consistent with stable compressor operation over a wide range of conditions, high component efficiency, minimum complexity and weight with allowance for engine structural requirements.

Apparently, the simplest solution would be to join up these end points with straight lines and fill them with the appropriate no of stages to do the pressure ratio and make allowance for structural struts to be suitably interposed. This however is not pratical in aerodynamic terms since compressor performance correlations dictate that falling line

compressors require to be lightly loaded aerodynamically at the inner wall and therefore many more stages than necessary would be required. Alternatively, a shorter, constant O.D compressor connected to the fan root by means of a Swan Necked Duct would allow the no. of stages to be drastically reduced (~12 stages for R = 20:1). Thus a cheaper, lighter engine would result. However, if this compressor were driven by a single shaft, a large degree of variable geometry would be required, in the form of variable geometry stators and bleed systems, to facilitate part speed operation without compromising the cruise operating efficiency. Rear stages restrict flow when insufficient pressure ratio is available to compress it. Therefore flow must be either bypassed through bleed ports and/or blade angles are adjusted to operate in low axial velocity flow in the front stages. The alternative to this kind of complexity is to use a two shaft compression system which allows the front and rear stages, otherwise on one shaft, to operate independently of each other.

In practice the range of manipulation offered by bleed system is far greater than variable geometry blade rows since the range of efficient variable geometry operation is limited. Fig 5 illustrates the no. of variables required as a function of pressure ratio and no. of stages.

Separation into two or more core compressor schedules also facilitates the separation of these respective turbines into separate modules which also have their stage matching problems. However a large disparity between L.P fan and L.P turbines can arise at very high bypass ratios. Essentially the L.P turbine mean diameter is related to the core size whereas the fan is not. Geared fans must be used for bypass ratios much above $\mu = 8$ in order to maintain a reasonable no. of LP turbine stages with optimised loading, see fig 6.

2. COMPRESSOR PERFORMANCE POTENTIAL

2.1 Stage Characteristics Compressor overall performance depends upon the behaviour of its constituent stages each of which have an inherent operating range determined by the basic vector triangle geometry associated with that stage, Fig 7. Fluid presented to the rotor at a preset flow angle acquires kinetic energy by change of angular momentum imparted to it by the rotor. The kinetic energy is subsequently recovered through diffusion by the stators. Unlimited response to a change in outlet throttle position assuming blade flow exit angles remain approximately constant would suggest a linear variation in work which is dependant primarily on the rotor outlet angle viz ideal slope in fig 7. However as the throttle opening is increased at constant speed a max flow capacity is reached when any further throttle increase merely increases losses. Reduction in throttle opening reduces the flow coefficient and work output is increased until the blade incidence becomes too large for blade suction surface boundary layer stability. At this point the losses increase rapidly and the characteristic slope passes through zero to a positive slope. When the slope becomes equal to that of a throttle characteristic, flow instability occurs and a dramatic flow reduction can occur which can amount to a complete flow reversal known as Surge - Fig 7A Alternatively the characteristic can operate stably with partial stall resident in the rotating cascade. This phenomona is known as rotating stall where the stalled zones reside stably along side unstalled zones, rotating at approximately half the compressor rotational speed. It is likely that these stalled zones tend to offload the remainder of the cascade by virtue of the blockage they create. Unstalled flow coefficients are locally increased. Thus a stable secondary characteristic can be operated upon by further closure of the outlet throttle until a sufficient no. of stalled zones reduce the total fluid momentum below that required to balance the static pressure gradient. This phenomenon represents a severe handicap to multistage compressors operating at part speed in the region of Ground Idle and below. Attendant inefficiency and flow reduction can inhibit acceleration to Take Off conditions. The front stages are driven towards stall by rear stage choking, see Fig 8. Principal course of action, taken to avoid this problem is by introduction of a bleed halfway along the compressor so that the front stages see an effectively larger throttle opening. Further action can be taken by skewing the stators to reduce blade incidences. The effect of this problem can be further reduced if front and rear stages are designed to cope with these off design excursions. If the rear stages are designed near to stall without loss of efficiency then an essentially larger flow range is afforded by the rear stages. At the same time if the front stages are designed with inherently more stall range then a better design compromise will be achieved, see fig 9.

Low hub/tip ratio fans do not present the same stability problem in spite of the inherent slope of the fan root characteristics associated with the conventional bypass engine - fig 10. Root characteristics have very little negative slope and would appear to have the propensity for surge. However, since no barrier exists between bypass and core streams then, automatic compensation for reduced flow coefficient is available by rematching of the split streamline. Incidence increase is reduced by movement of the split streamline at inlet which although it tends to create an adverse area ratio across the fan root, improves the area ratio of the flow adjacent to it. Thus the root flow is stabilised by the bypass flow regime. The bypass portion of the fan is not subject to the same rigours as a multistage simply by virtue of the lower level of pressure ratio, see fig 8.

2.2 <u>Overall Characteristics</u> The performance of a typical multistage compressor, designed for operation in a typical civil application, must be capable of stable efficient operation when subjected to a wide variety of performance deteriorating pressures. As well as coping with its inherent off-design limitations, rotating stall, surge and choke, it must be able to absorbe a range of environmental effects. Flow distortion caused by cross winds, upstream spoiling by various components (struts, proceeding compressor profile) can seriously deteriorate the surge line to the point where it impinges on the working line if insufficient surge margin is built into the design. Tip clearance effects can equally degrade the surge line and the total pressure profile which enters the combustion system. Adeficit in tip total pressure at combustion entry reduces the tip flow entering the combustor and deteriorates the mixing process. Mechanical design to overcome this problem is difficult since it depends on the relative thermal stiffness of rotor casing and disc. The problem can be particularly acute at Take Off conditions immediately after acceleration from Ground Idle. Light-weight compressor casing expands rapidly to create a rotor-casing gap while the rotor and disc suffer a much greater thermal growth inertia. The disc having a much larger mass per unit surface area responds to temperature increase more slowly. Further surge capacity is required to ensure that during rapid accelerations the transient working line does not impinge on the worst case surge line - Fig 11. A rapid rise in combustion pressure, due to fuel input, effectively throttles the compressor at constant speed. Not until the turbines respond to this excess energy input can the compressor accelerate to match the demand. Deceleration does not present this problem. For a typical 5 or 6 stage compressor something approaching 40% surge margin is required to absorbe the total effects of environmental surge line deterioration and transient working lines. In practice, the combustion fuel input is scheduled with speed so that the potential surge problem is avoided.

Twin spool compressors appear to lend themselves to a more efficient automatic response to this effect than do single spool core compression systems. The single spool device is more cumbersome in that several variables have to be manipulated during acceleration probably with the aid of computer control. Mechanically, variables have a habit of requiring heavy mechanisms to operate them against the blade forces in the compressor. Twin spool core compression systems do not have this complication since it is usually the second stage or High Pressure compressor which is most likely to surge. At ground idle the HP compressor is running efficiently at about 80% of the Take off speed, well clear of the surge line which the first stage or Intermediate Pressure compressor is idling at 40% matched to its own conditions. The HP therefore protects the IP both of which reach 100% Take Off speed automatically surge free. The single spool compressor however is plagued with inefficiency owing to the extreme degree of variable stator manipulation. Also the whole mass of the compressor has to be accelerated with greater inertia than in the twin spool device. Fig 12 illustrates the difference in behaviour between the two.

Fan performance does not suffer such a variety of performance degrading effects except for inlet flow distortion. The fan is most vulnerable to this since it is in the direct line of fire due to cross winds. These represent the greatest hazard at T.O., when the aircraft is stationary. However, as soon as significant forward speed is attained the intake generally clears up when crosswind velocities become insignificant compared to forward speed. It has been demonstrated, however, that with suitable fan tip treatment (skewed slots) this effect can be significantly reduced. Tip clearance does not represent a problem due to the low hub/tip ratio of the fan. Transient working lines do not represent a major problem since the only condition where they apply is on landing when the thrust reverser, located in the bypass duct is actuated. Transiently the bypass plenum pressure approaches stagnation until the thrust reverser is fully open. Rapid operation of such a device usually avoides the problem.

Fan root stability is largely influenced by excursions to extreme values of bypass ratio. At Flight Idle bypass ratios reaching up to 15-20 relative to design values of between 5- 8 can introduce extreme flow curvature in the root section which could influence the annulus wall boundary larger stability - Fig 13. However part speed operation is automatically dealt with by spillage over the bypass splitter.

Having observed that fan bypass surge margin at Take off to be a potential hazard it can be seen that the sea level static working line diverges from the altitude cruise working line more considerably as the pressure ratio is reduced. If the compressor is designed at cruise for maximum operating efficiency then it becomes apparent that a large design surge margin must be obtained to cover the wide range of operation between cruise and T.O at S.L.S working line - see fig 14. Design for T.O conditions would merely penalise the max obtainable operating efficiency at cruise. This represents a considerable design problem which potentially could be overcome with variable geometry rotors - very cumbersome. Alternatively if the pressure ratio was low enough and the engine size large enough it may be worthwhile using low aspect ratio hollow fan blades - Aspect ratio having a strong effect on surge potential. At small engine sizes the hollow fan is not practical since the blade skin thickness becomes comparable with the total blade thickness which introduces a large weight penalty.

Fig 15 summarises the state of the art in terms of achievable efficiencies per stage. Significant progress has been made over the last decade which enable higher pressure ratios to be achieved at efficiencies that have been acceptable for engine performance. Principal improvements are associated with improved knowledge of how to design for high Mach No. blading consistent with the higher levels of pressure ratio

and the adoption of differing types of design vector triangle geometry suitable for higher pressure ratios. Accumulation of data and knowledge of blade diffusion limitations in conjunction with conscious design which accounts for secondary flows and radial variation of loss have also been of benefit. Recent testing of more advanced transonic designs at higher pressure ratios suggest that the standard of technology over the next decade is likely to improve at similar rate to that experienced over the previous decade. However, in order to realise this potential, compatible progress in mechanical design must evolve so that higher mechanical blade speeds, consistent with higher pressure ratios, can exist in an engine environment satisfactorily.

3.0 BLADE AERODYNAMIC PERFORMANCE

3.1 Free Stream Blading In order to predict compressor performance it is necessary to describe the behaviour of individual blades and to know their limitations. Fig 16 illustrates the principle parameters which describe blade performance in the free stream sections of the blade remote from the secondary affects of other fluid boundaries (end walls). Blade forces acting on the fluid to produce a change in angular momentum and diffusion create a flow deflection relative to the blades proportional to the magnitude and distribution of the lift forces exerted by the blade. The mass mean flow direction is usually at variance with the blade outlet angle producing a flow deviation. The magnitude and distribution of this lift force is largely dependant on the blade geometry in terms of blade angles, solidity and blade shape. Aerodynamic loading in terms of the standard Lieblein Diffusion Factor, Axial Velocity Ratio and level of Mach No tend to act in a secondary way by affecting the distribution of deflecting forces-lift.

Losses are dependent upon the blade boundary layer existing which is controlled by level of Diffusion, distribution and level of M_N along the blade surfaces and the impact of shocks set up within the blade passage. Contrary to flow deviation, losses are strongly determined by flow incidence to the blade inlet, as well as the level of MN in the passage throat Fig 16. Depending on the level of M_N at inlet and the critical area ratio or choke margin, a subsonic blade rapidly chokes with reduction of incidence and flow reduction results. Observations of the static pressure field generated at various conditions from max flow coefficient to near surge reveals the propensity for choke and stall - fig 17. At low incidence near max flow conditions the peak velocity occurs in the region of the subsonic cascade throat with resulting high static pressure gradient downstream of the peak on the suction surface. Further reduction of incidence would increase this gradient to the point of B.L separation and the resulting blade passage blockage would restrict the flow at sonic conditions. Increase of incidence relieves the static pressure gradient in the latter half of the blade and imposes it near the leading edge. It is likely that leading edge stall flow reattaches due to turbulent mixing. If the entry MN is high enough then reduction of incidence is likely to create suction surface shock which separates the boundary layer - Fig 18. Cascades operating with entry Mach Nos. of 0-8 and above will generally be supercritical at low incidences with a normal shock extending over the whole of the blade passage. It is therefore feasable to anticipate a loss model based on a Diffusion parameter and passage M_N or choke margin - fig 19.

Blade sections normally used for multistage applications range from the British C4 which is really only suitable for operation below entry M_N of 0.5. As the level of MN reaches 0.7 it becomes critical at nominal minimum loss incidence and therefore any significant range of operation can only be obtained below MN 0.5. High curvature in the forward arc, with max thickness at 30% of chord from leading edge dictates the level of critical M_N. Reduction of the curvature can be achieved by recourse to Double Circular Arc type blades, which can operate, at nominal incidence between $M_N = 0.7$-1.2 depending on the degree of blade camber. Characteristically stator vanes have high camber ($\sim 50^o$) and the critical M_N is reached at nominal incidence, when inlet $M_N \approx 0.9$. D.C.A sections only operate satisfactorily at MN $\approx$ 1.2 if camber is less than 10^o. For M_N's in excess of 1.2 recourse to Multiple Circular Arc sections and variants there of are used in the tips of transonic fans. These are designed to minimise the Suction Surface M_N to reduce the impact of leading edge bow shock when it strikes the leading blade suction surface. The criticality of inlet M_N is obviously no longer an issue.

The function of a transonic/supersonic inlet blade typical of that found in the tips of transonic fans is to not only provide a change to angular momentum but has the more complex task of supersonic diffusion as well as subsonic diffusion within the blade passage. In general compressor designs have not succeeded in doing this without strong shock losses being present. The attempt is made to supersonically diffuse up to the passage throat downstream of which the flow should diffuse subsonically, but in practice the flow tends to reexpand supersonically again only to achieve the appropriate subsonic downstream conditions through a strong normal shock. Witness the static pressure field measured by fan tip transducers in Fig 20. A possible flow model which could describe this process is represented in Fig 21. However it would appear that this kind of flow model is more complex since attempted simulation of the blade passage aerodynamics has by no means achieved generality. Loss predictions however can still be made at ideal conditions with a crude degree of accuracy by attributing loss to normal shocks associated with the mean passage entry M_N and the general level of diffusion taking place. The level of choke margin that the blade passage operates at has been found to strongly determine the level of passage shock loss and more generally the off-design performance of a transonic fan. Fig 22 illustrates how although high efficiency can be achieved at the design point by use of small choke margins in supersonic flow, some penalty in flow and part speed peak efficiency transpires. High passage Mach Nos. persist at lower speeds due to area ratio

effects and therefore high blade passage loss reduces efficiency and max flow achievable.

3.2 Secondary Flow Effects The effect of secondary flows on compressor performance may be as profound if not more so than the free stream aerodynamic behaviour. Typically high losses are generated at blade end wall interfaces where three dimensional diffusion tends to generate large corner vortices and local separations. In a compressor design increased work at the blade ends is usually injected to counter the strong losses at the blade ends, rather than off loading the ends to relieve secondary loss generation. Fig 23 illustrates the type of end wall secondary loss growth four stages into a highly loaded multistage. Almost 20% of the annulus at each end is submerged in boundary layer.

Note that rotor cascade performance cannot be predicted by stator cascade performance because of the way in which each type of blade end sees the secondary flow. The stator cascade operates in the absolute frame of reference and consequently is affected by the end wall boundary directly. The rotating cascade however only responds to this secondary flow growth transiently and sees it as a locally very high dynamic head flow due to the low axial velocity. Thus rotary cascades will always have greater performance range them stators. This factor can be used to advantage by increasing the rotor loading using reaction design to off load the stator. Fig 24 illustrates a typical stator exit pressure profile which shows how the end wall losses interact with the free stream losses. The static pressure field generated in the free stream part of the blade is imposed on the low velocity flow at the end wall which tends to create overturningof the low velocity flow. A further effect of secondary loss generation in stators comes about due to the inherent static pressure gradient which causes tip secondary flows to drain radially inwards. This not only thickens free stream blade surface boundary layers but also feeds the root which is usually more highly loaded than the tip. In rotors however the reverse is likely to be true since low axial momentum flow acquires high whirl velocities at rotor exit than the free stream value. Therefore the secondary flows will drain radially outwards to the tip. This feature is responsible for a significant loss of efficiency in rotor tips due to thickening of the boundary layer and particularly penalises transonic fan tips.

Blade aspect ratio is a strong determinant of stage performance by virtue of the effect it has on secondary loss generation - Figs 25 and 26 show the effect on surge margin and efficiency respectively. It is probable that at high aspect ratio the relative secondary losses become a larger proportion of the local passage flow due to changing Reynolds No based on the blade chord or passage width. The viscous term becomes more predominant relative to the momentum term ($\rho U x / \mu$). Furthermore the effect on free streameffective area ratio due to end wall blockage becomes less compensatory. In low aspect ratio blading having long chords and large passage widths the fluid momentum term is increased relatively and the resulting blockage may be more progressively distributed in such a way as to improve the free stream area ratio. The off-loaded free stream will therefore tend to stabilise the performance of the total blade row.

Surge Margin capacity is probably limited due to the area of low velocity flow acting against the free stream static pressure gradient. Closure of the outlet throttle increases the disparity between fluid momentum in the secondary flow require and the stator pressure gradient, up to the point where reverse flow takes place. Hence surge may emanate from the secondary flow regions of the blade.

3,3 Environmental Effects The axial compressor has been shown to be vulnerable to a variety of environmental effects which can seriously impair the performance at critical operating conditions:-

(i) Flow distortion in the form of total pressure variation in a field of approx. constant static pressure, regionally adjusts the blade flow coefficients such that compressor instability occurs prematurely. Blades passing through the region of reduced flow respond transiently to influence of reduced flow velocity. Initially each blade tends to be off loaded due to the reduced axial velocity ratio initially. However, as the blade progresses through the low velocity flow, this situation returns to normal AVR values at reduced velocity levels. Finally as the blade leaves the low velocity zone increased inlet velocity with resident low exit velocity increases the diffusion load above that experienced normally. Then, not also does a transient diffusion increase occur but also closer proximity to surge by virtue of the effect of reduced axial velocity onvector triangle behaviour. Spoilt and unspoilt regions of the rotor act in parallel to produce a reduced average range of stability for the total compressor - Fig 27. Progression of the spoilt zone through a compressor tends to be attenuated however within three or four stages depending on the constituent stage characteristics. Reduced axial velocity will in general cause higher total pressure to be generated which will offset the total pressure deficit at outlet. However, in consequence to this, it is inevitable that total temperature distortion will be produced which is likely to influence a downstream compressor by changing regional Mach No distribution. The above argument tends to apply to circumferential spoiling only and radial spoiling tends to act in a non-transient manner if it is fully circumferential. Tip or root spoiling will locally adjust operating incidences and flow coefficients which give rise to a redistribution of radial work.

It has been shown that only a nominally small degree of circumferential spoiling is necessary to achieve the full effect of spoiling - Fig 28. This is due to the transient nature of the blades response to passage through successive low and then high axial velocity on exit from the spoilt region. Apparently only 60° - 90° spoiling is necessary to achieve the full effect depending on the transit time of flow within the blade passage.

(ii) Reynolds No effects give rise to changes in performance at various engine flight conditions and must be accounted for when relating rig performance at I.S.A conditions to engine size components. Fig 30 expresses this effect on Surge Pressure Ratio and Efficiency relative to Re. nos. based on inlet axial velocity and compressor annulus height. This effect will obviously vary for compressors of different stage nos. due to large internal variation of Reynolds No. It is probably more relevant to express performance changes in terms of mean compressor Reynolds Nos. or even account for individual blade row performance in these terms.

4.0 DESIGN OPTIMISATION

4.1 Multistage Design In order to maximise the potential for efficiency, without sacrificing basic stability requirements, each compressor stage must be optimised in terms of loading consistent with the blading rules (solidity) applied. This is most conveniently expressed in terms of the balance between shaft horsepower input and work done by the compressor blade rows expressed non-dimensionally as $\Delta H/U^2$ in relation to Va/U, functionally related by vector triangle geometry. Fig 31 enables these factors to be translated into terms of static pressure rise coefficient and level of Mach No dependent upon the level of pressure ratio being considered. Therefore the potential exists for loss prediction based on some diffusion parameter and a suitable shock loss model. Rotor and stator diffusion and M_N level can be optimised to achieve a balance appropriate to the level of pressure ratio being designed for. Parametric study, for a range of pressure ratios, indicates that a peak efficiency point can be obtained due to the limit of diffusion. This is consistent with the behaviour of experimental diffusers which progressively approach minimum loss and then proceed to stall rapidly as the level of diffusion is further increased.

This type of procedure can be studied for a range of stage reactions to further refine the process. Reaction defines the proportion of static pressure rise done by the rotor in relation to that of the stage. Fig 32 shows the various vector triangle geometries associated with different levels of reaction. 0% reaction corresponds to a simple Pelton wheel type arrangement in which no static pressure rise is done by the rotor. However this type of design would not be capable of significant pressure ratio due to the low dynamic head seen by the rotor. Also stator M_N is very high. The vector triangles are shown for constant axial velocity but some rearrangement to more favourably dispose the loading and M_N could be achieved by change of AVR. 50% reaction distributes the diffusion rates between rotor and stator equally but an upper limit to pressure ratio achievable will be reached when the resulting stator entry Mach No exceeds 0-8.

Higher pressure ratios and simpler design can be achieved by use of the zero α_0 type of vector triangle geometry, due to higher rotor relative dynamic head availability. There is no requirement for an inlet guide vane to direct the flow on to the rotor at a suitable angle and the stator deflection requirement is further reduced. This type of design is suitable for pressure ratios between 1.5-2.0. If further increase in pressure ratio is required the reaction must be increased up to and beyond 100% so that suitable levels of dynamic head are available to maintain diffusion levels within the bounds of practicality. Also stator entry M_N's can be maintained at values less than critical consistent with the degree of stator blade camber.

Axial loading distribution is optimised in such a way as to obtain the best compromise between design and part speed operation. If the rear stages are too lightly loaded then high stagger rotors tend to reduce the available choke margin at the compressor exit. This is likely to create heavy stall in the front stages at part speed conditions. Alternatively high pressure ratios in the rear stages will reduce blade staggers and effectively provide for greater choke margin in the rear stages. However blade stall margin requirements will inhibit the use of excessive pressure ratio in the rear stages. Also, characteristic slopes will be reduced which are likely to predispose the compressor to reduced surge margin. The rear stages control surge margin at high speeds, but if the work level is already high then higher design speed pressure ratio achievement with reduced surge margin may produce a nett result which is similar to that obtained using lightly loaded rear stages.

If high reaction front stages are used in conjunction with low reaction rear stages of a multistage compressor then a better part speed compromise can be achieved. High reaction in the front stages is more suitable for the generation of high pressure ratios which inherently relate to the front half of a multistage. Low reaction (50%) is more suited to lower pressure ratios found in the rear stages. [illegible], the reduction in near stage rotor staggers associated with low reaction gives the rear stages a wider choke range. This facilitates the passage of larger exit flows at part speed and therefore reduces the degree of flow reduction associated with the front stages at part speed. Having optimised the velocity triangles and distributed the pressure ratio to max. advantage blade solidity is chosen to maximise the off-design

potential. Cascade test results are referred to and quantified in terms of $\alpha - M_N$ diagram which relate the available stall and choke margin, the boundaries of which are defined by the incidence at which twice minimum loss is obtained. Fig 33.

If the facility for blade blockage prediction is not available, some allowance in the vector triangle geometry must be made for the accumulation of wake blockage. This can be done in one of two ways. Either an empirically corellated blockage factor can be used to elevate the level of axial velocity in the annulus or increased work can be injected into the geometry to compensate for work lost through operation at higher than design axial velocities. Blockage factor design is probably more realistic, provided an accurate knowledge of blockage can be predicted. However work done factor designs is likely to be safer since extra work is predesigned to cater for a wider margin of error - Fig 34.

4.2 Transonic Fan Design Transonic fans are designed to max. Diffusion Factor limitations in conjunction with vector triangle geometry tuned to optimise choke margin for max efficiency related to previous discussion of passage aerodynamics. Blade shape is tailored to the particular environment with due attention to suction surface MN control. Owing to the widely varying nature of the blade aerodynamics. Blade speed varies considerably over typically low hub/tip ratio fans (0.3 - 0.4). At the root the tendency to create an impulse type of blade relates to the very low blade speed. This root section can therefore run into critical Mach No problems. Fig 35 illustrates the effect of uniform increase in pressure ratios at a given blade speed. The natural tendency for air deflection to reduce as blade speed increases fortunately aids the ability of blades to operate with low suction surface M_N's but care in design is necessary to avoid overloading in this respect.

Choke Margin is calculated on the basis shown in Fig 36 assuming appropriate distribution of loss and work done due to radius change.

Blade solidity can be determined by calculation of blade surface pressure coefficients downstream of the point of max. suction surface M_N provided accurate knowledge of boundary layer development up to this point can be computed. A typical diffuser loading criterion then determines the equivalent blade surface length required to maintain attached flow - see fig 37.

5.0 CONCLUSIONS

5.1 Development Areas

(i) Performance prediction for a given set of conditions depends firstly on the ability to simulate compressor flow field requirements and secondly, to predict how compressor blades are likely to respond to that environment. Flow field prediction readily computes vector geometry and the effects of curvature but usually fails to simulate boundary layer effects, especially in high curvature environments, and the effects of blockage induced by blade stall. Blade row performance is still largely determined by 'black box' type correllations and various attempts to include intra blade row computation have not been successful because the models used are usually oversimplified. Generalised prediction methods are still some distance in the future owing to the complexity and variety of blade aerodynamics.

Flow and pressure ratio at a speed depend on the ability of the blades used to generate lift and correllations are available to predict deflection for a wide variety of blade shapes.

Efficiency prediction rests on the correlation of diffusion parameters related to choke margin and various simple flow models can describe performance for related blade types.

Stability range prediction relates gross mean compressor loadings and blade geometry to achievable surge margin. The mechanism of surge remains ever elusive.

(ii) Off-Design performance predictions rests not only on an understanding of blade row performance but also on the nature of secondary flow effects. So many non-linear interactions occur that have a habit of defying generality. Some success has been obtained however using the simple 'Stage Stacking' method which feeds on the test data of a given compressor until a satisfactory mean line overall result is obtained. Unfortunately this type of analysis represents an amalgam of extraneous effects which can obsure the real blade behaviour.

The principal difficulty with off-design prediction of compressor performance is related to the anticipation of blade stall and the consequent effects.

Each aerodynamic section has its own limitations which interact with other blade sections. It is likely that blades can only achieve a max level of diffusion before the incursion of loss and blockage with more extreme

incidences actually reduces the real diffusion level. Stall is likely to cause flow redistribution along the blade and consequently adjust the loadings of adjacent blade sections. Off-design performance prediction is notable for its lack of success.

(iii) Data Acquisition lies at the root of blade aerodynamic understanding. A step change in the level of technology associated with data acquisition is required in order to make significant inroads into the fog of empiricism associated with compressor technology. Recourse to lasar anemometry appears to be hopeful, particularly in the field of rotating compressor hardware. However the associated computer power and manpower requires some considerable investment.

5.2 Future Potential With the advancement of fundamental aerodynamic understanding it should be possible to make significant advances in compressor technology which simplify the end product. It is likely that the description of off-design compressor performance will become more generalised and facilitate the design of multi-variable type compressors capable of use in variable cycle type engines aimed at low energy consumption and greater flexibility of aircraft use.

The description of supersonic blade passage flow and the trick of designing single stage compressors with pressure ratios between 5 and 10:1 may well be in sight.

Finally it is worth mentioning that some benefit may be obtainable by back-tracking technology to the point when engine components do not sit at the extremes of their respective technologies in order to simplify and cheapen consumer type transportation.

NOTE: It will be noted that Figure 29 is missing from the Figure sequence.

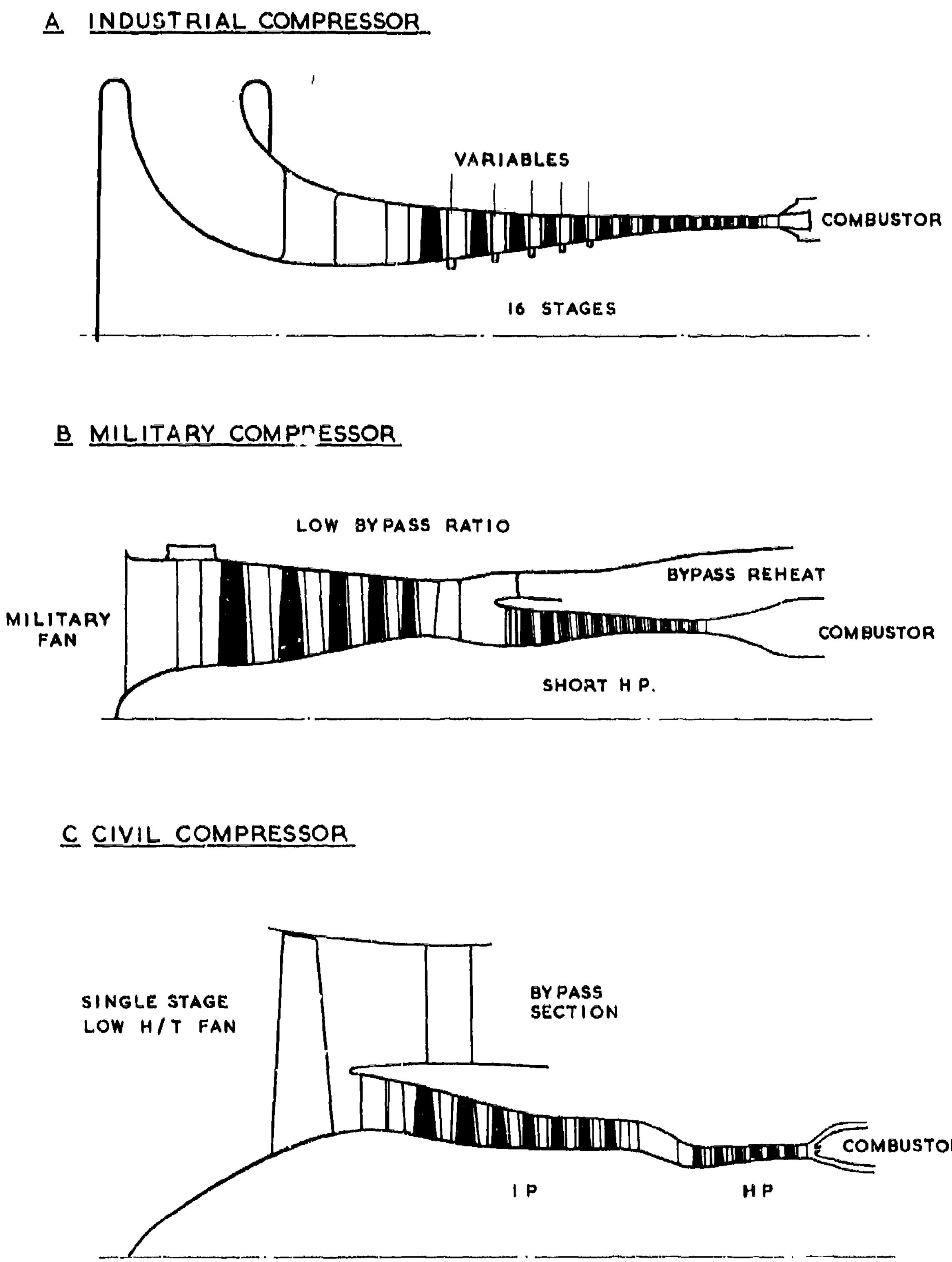


Fig.1 Axial compressor applications

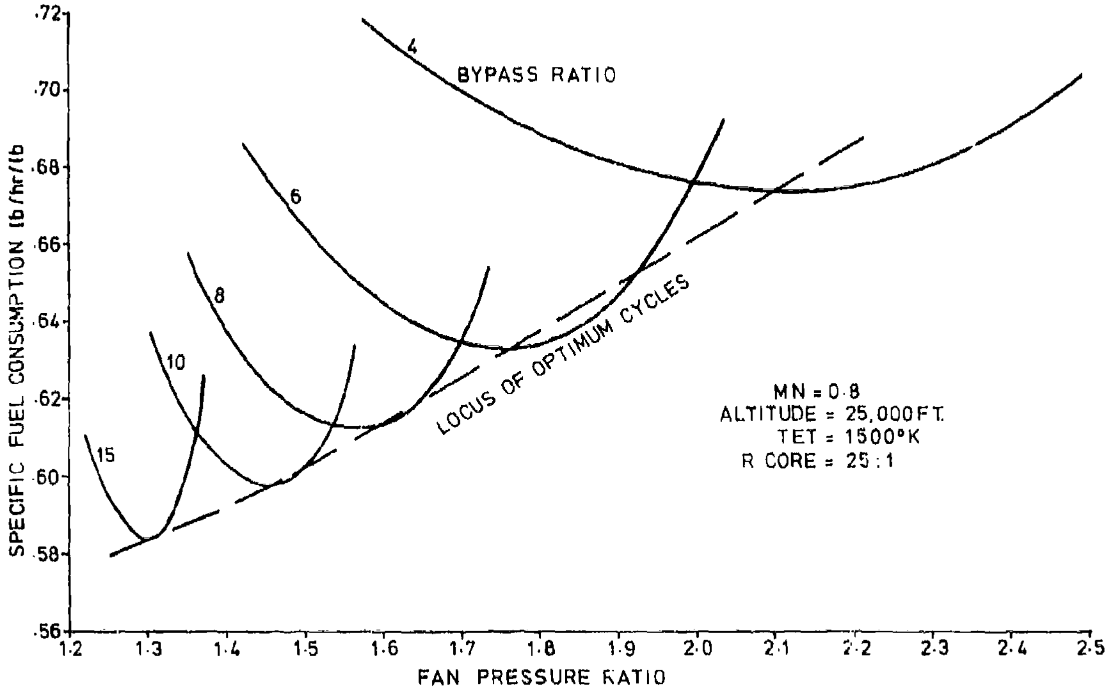


Fig.2 Effect of bypass fan pressure ratio on S.F.C.

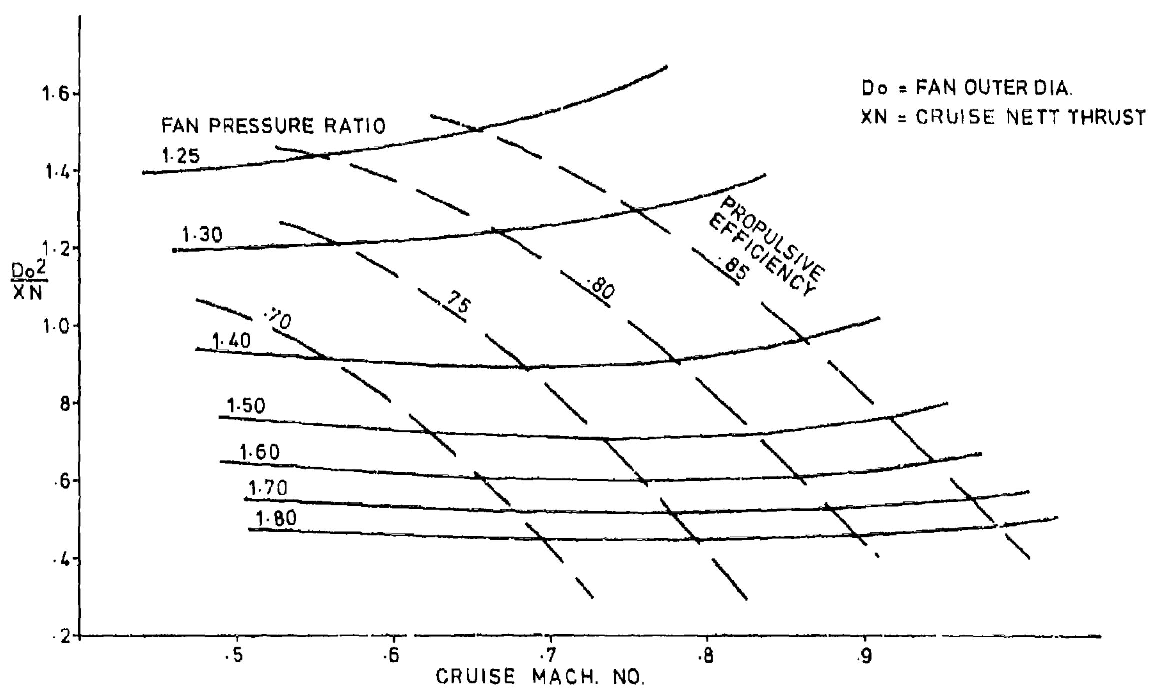


Fig.3 Effect of fan pressure ratio on propulsive efficiency

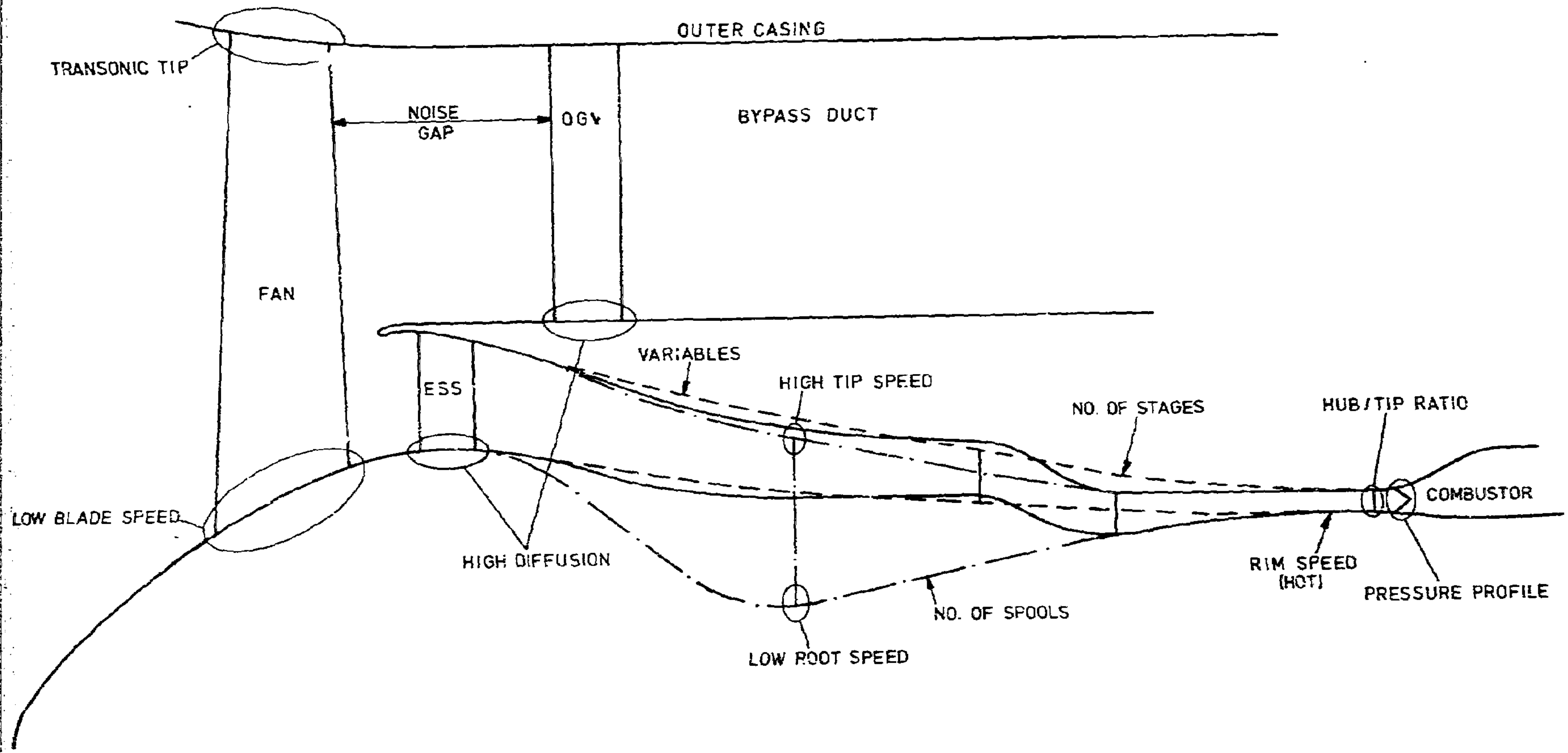


Fig.4 Bypass engine alternative configurations

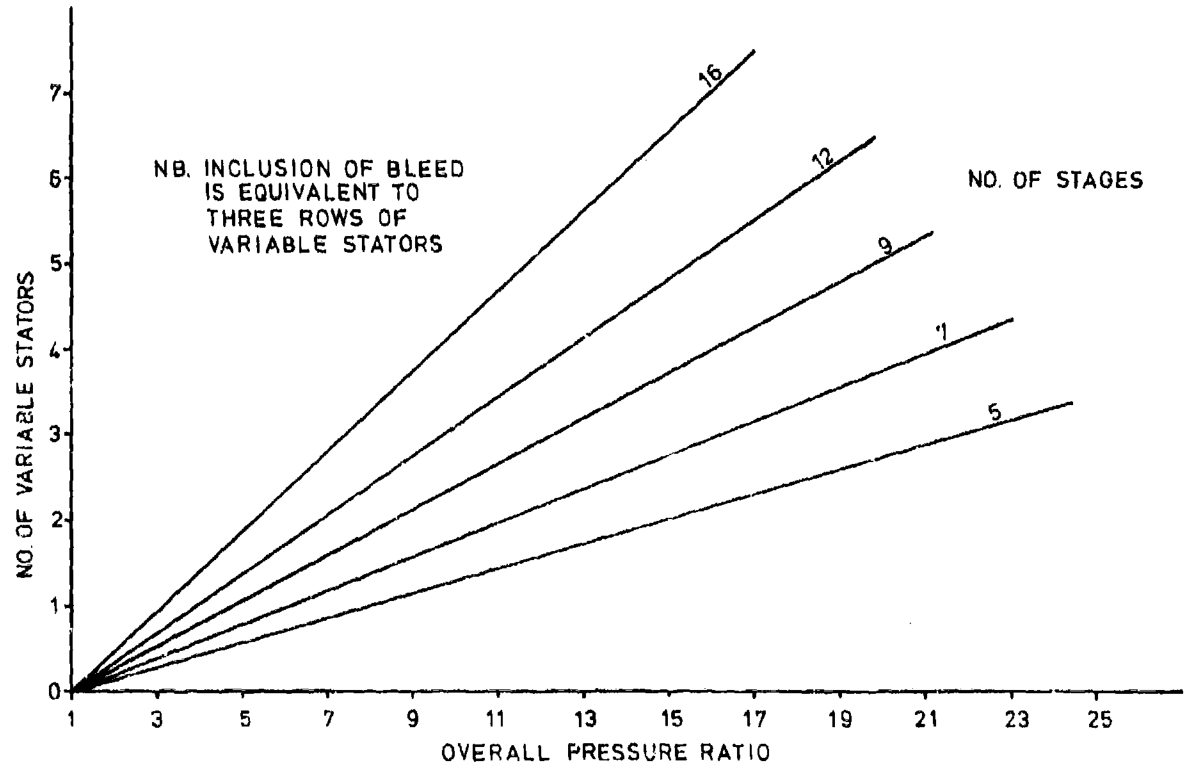


Fig.5 Number of variable stators per spool

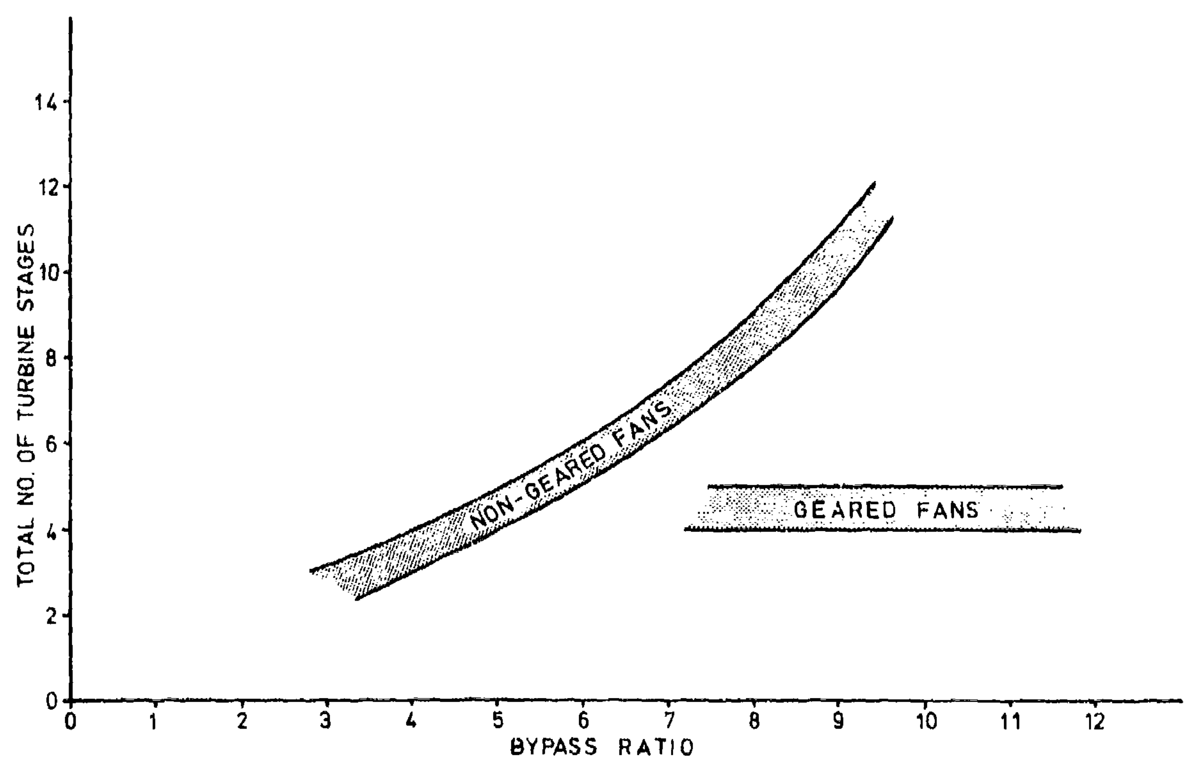
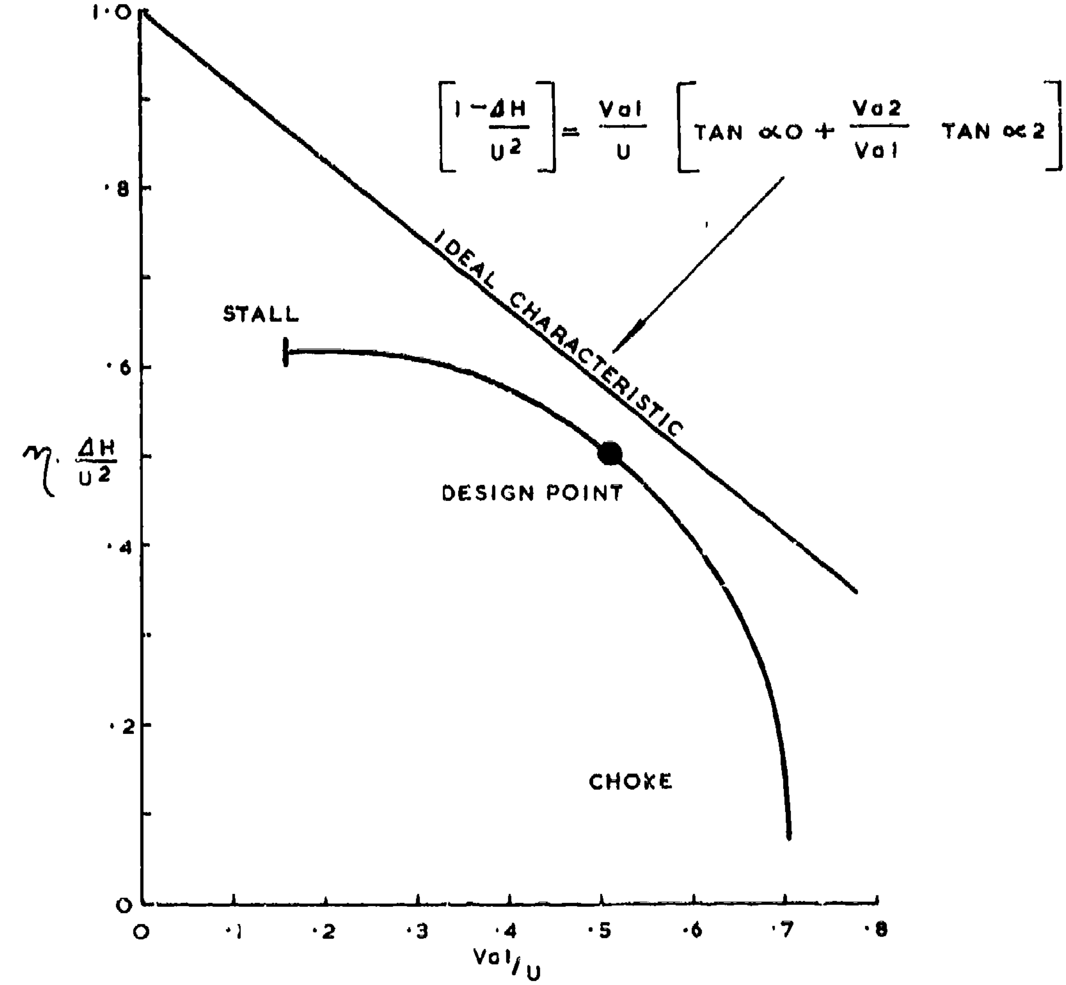


Fig.6 Effect of bypass ratio on number of turbine stages



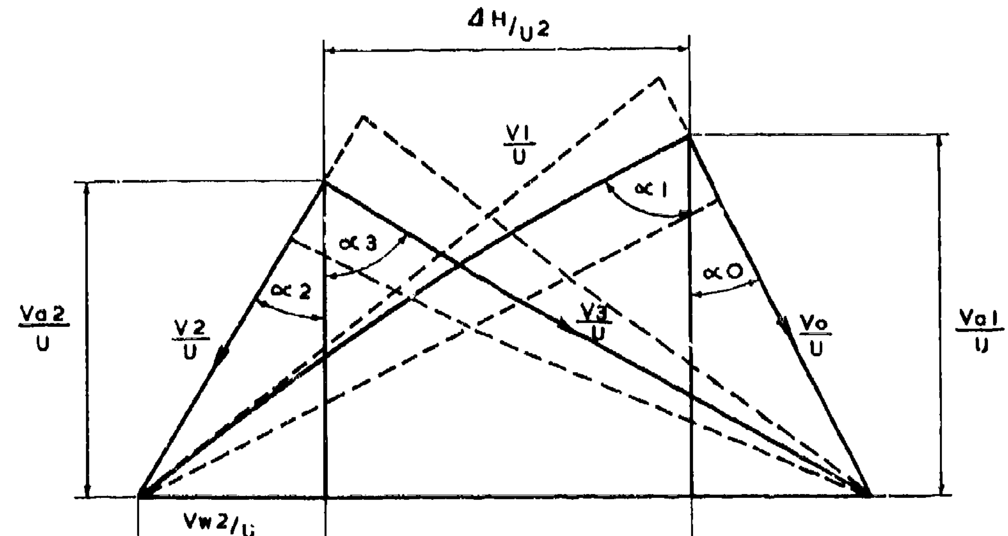


Fig.7 Stage characteristic performance

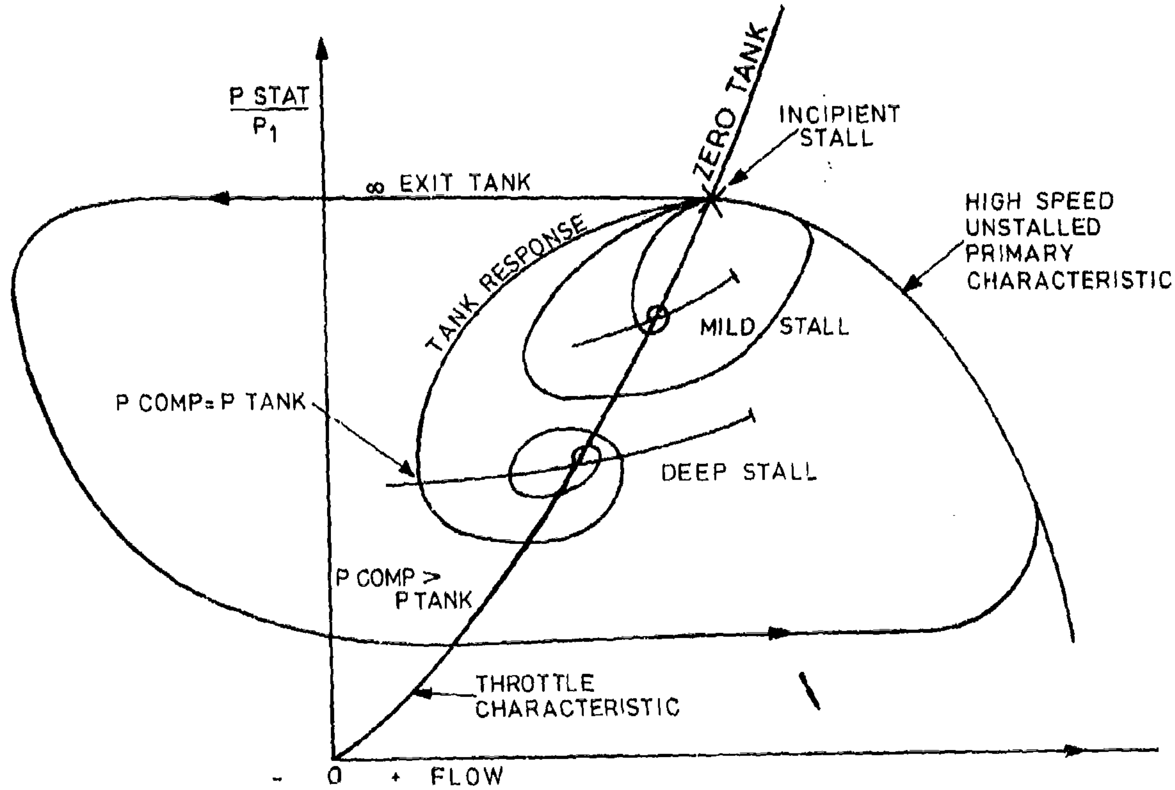


Fig.7A Compressor surge characteristics

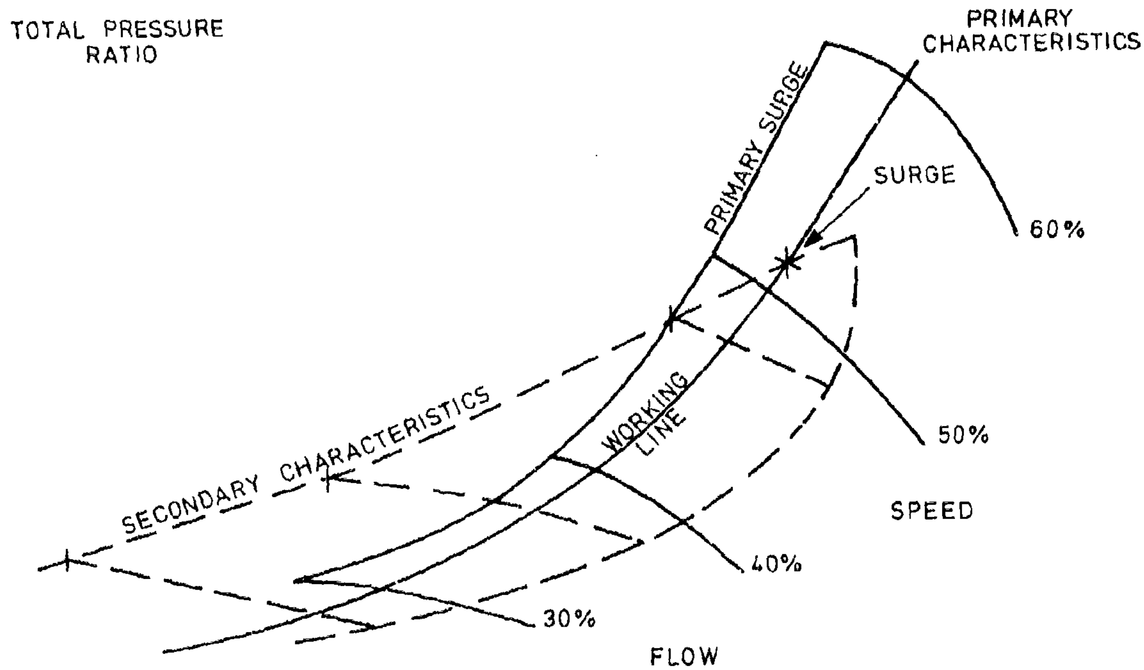


Fig.7B Compressor secondary characteristics

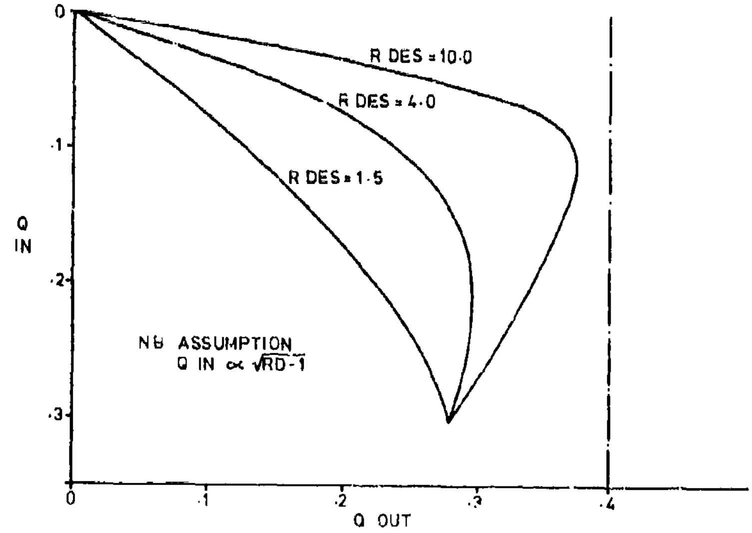


Fig.8 Component flow ratio at off design

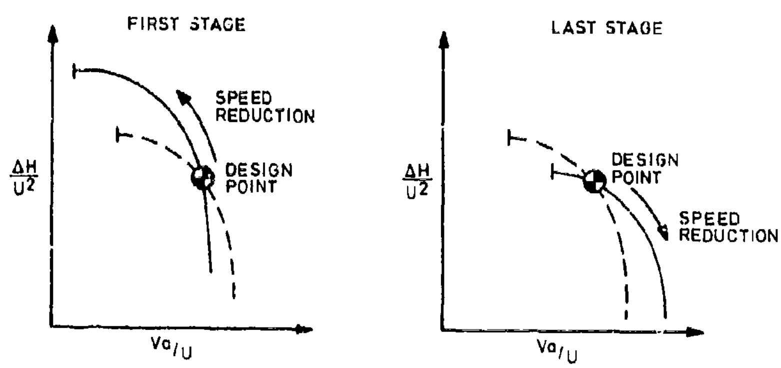


Fig.9 Multistage characteristics for first/last stages

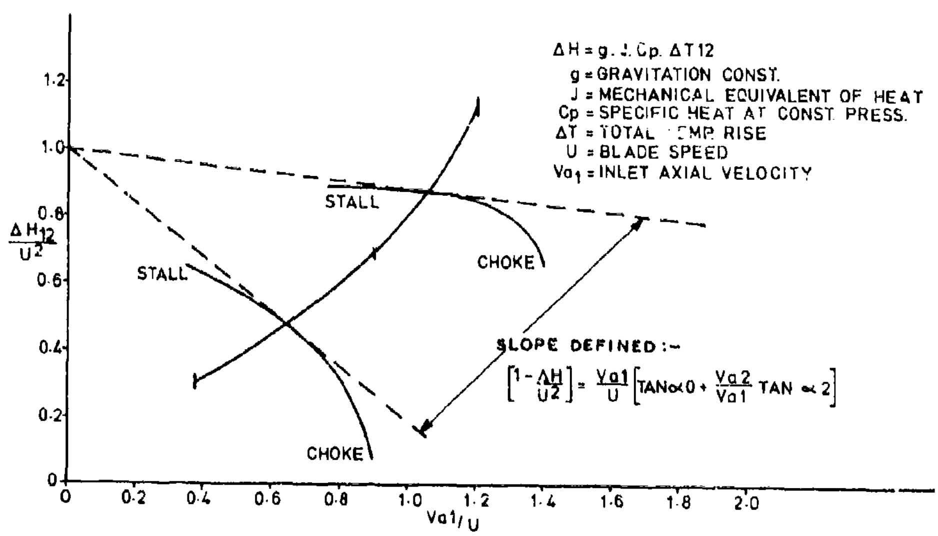


Fig.10 Root and tip characteristics for low hub/tip fan

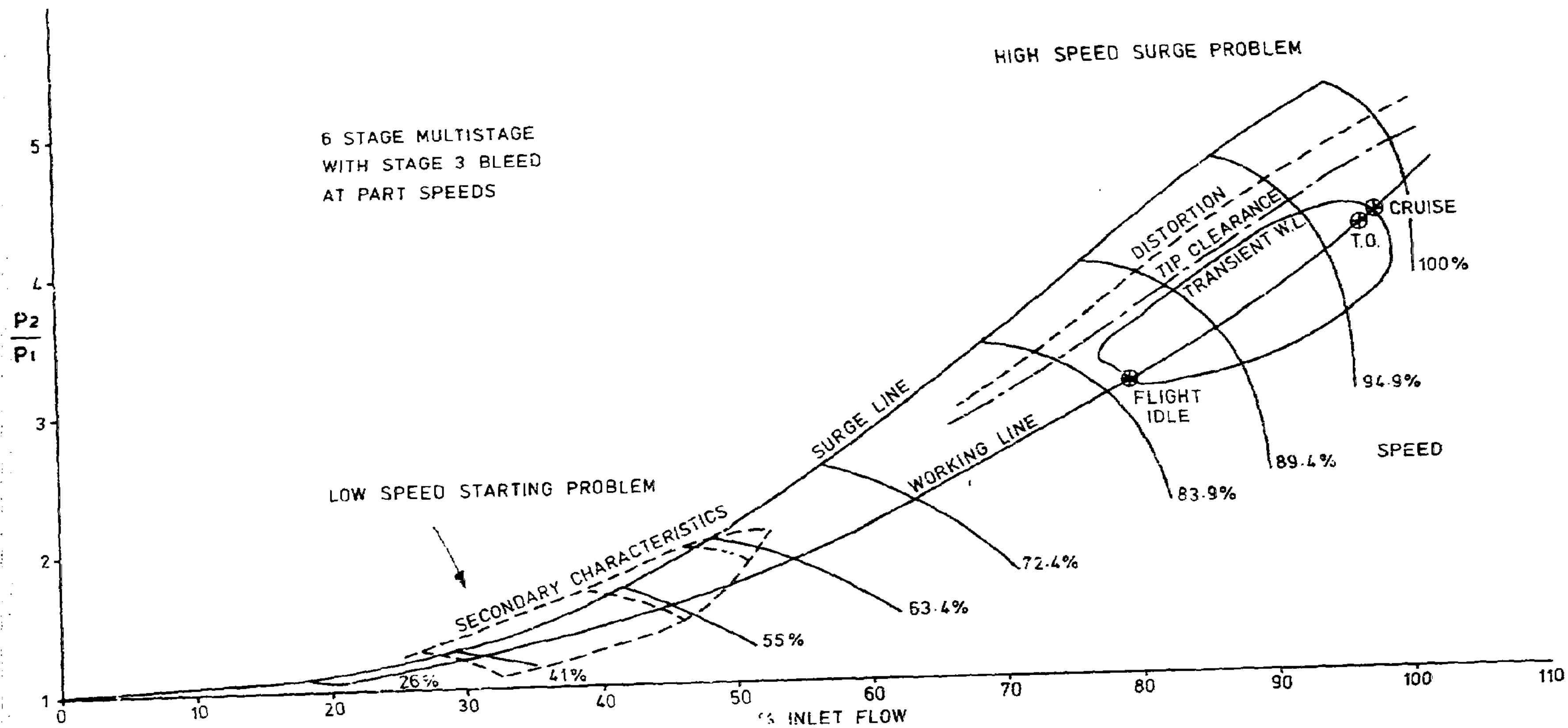


Fig.11 Multistage overall performance

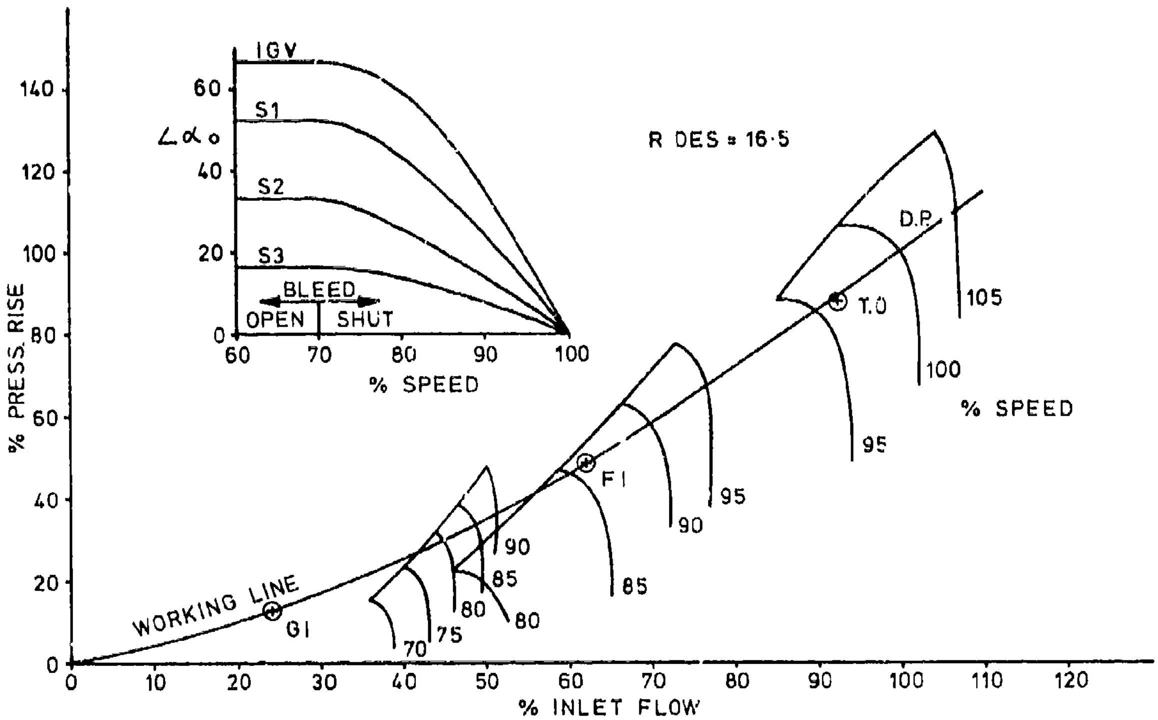


Fig.12A Performance of a single spool multistage

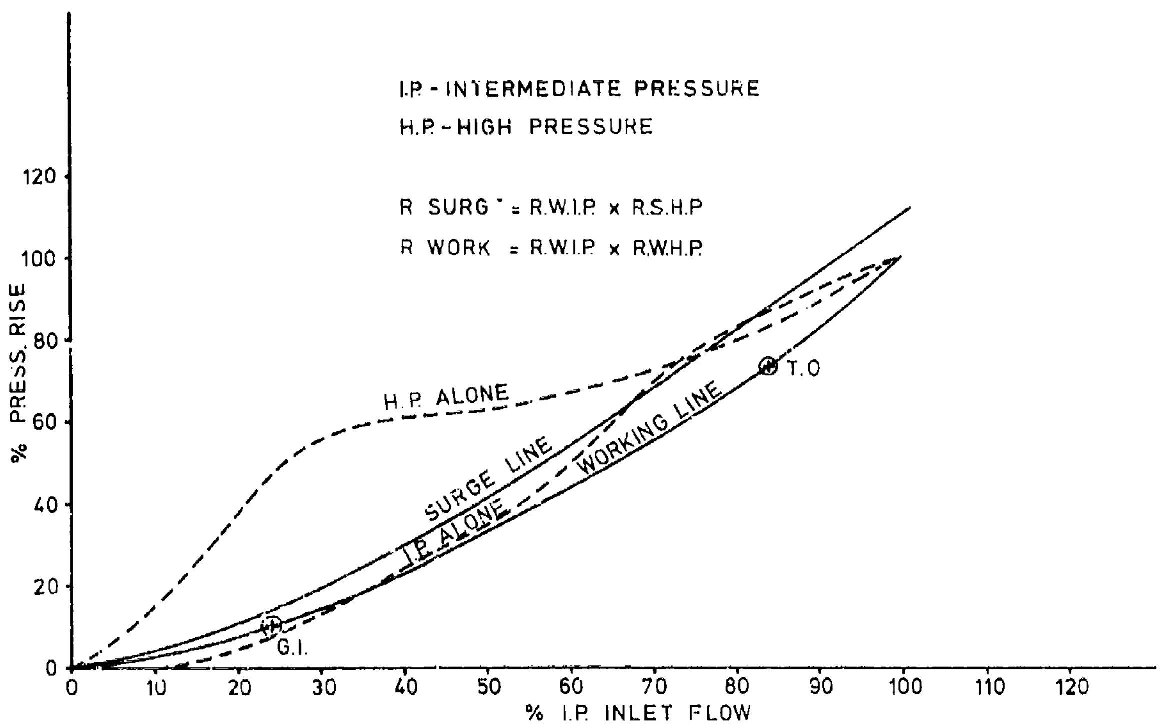


Fig.12B Performance of a twin spool multistage

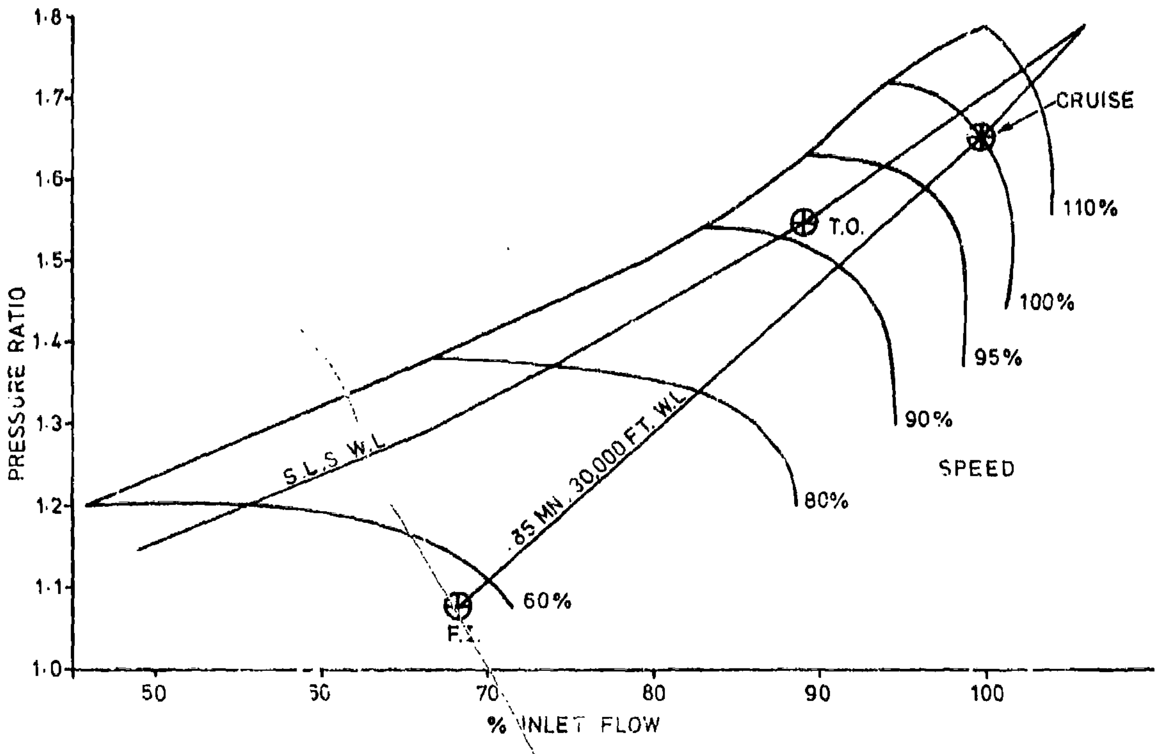


Fig.13 Bypass fan overall performance

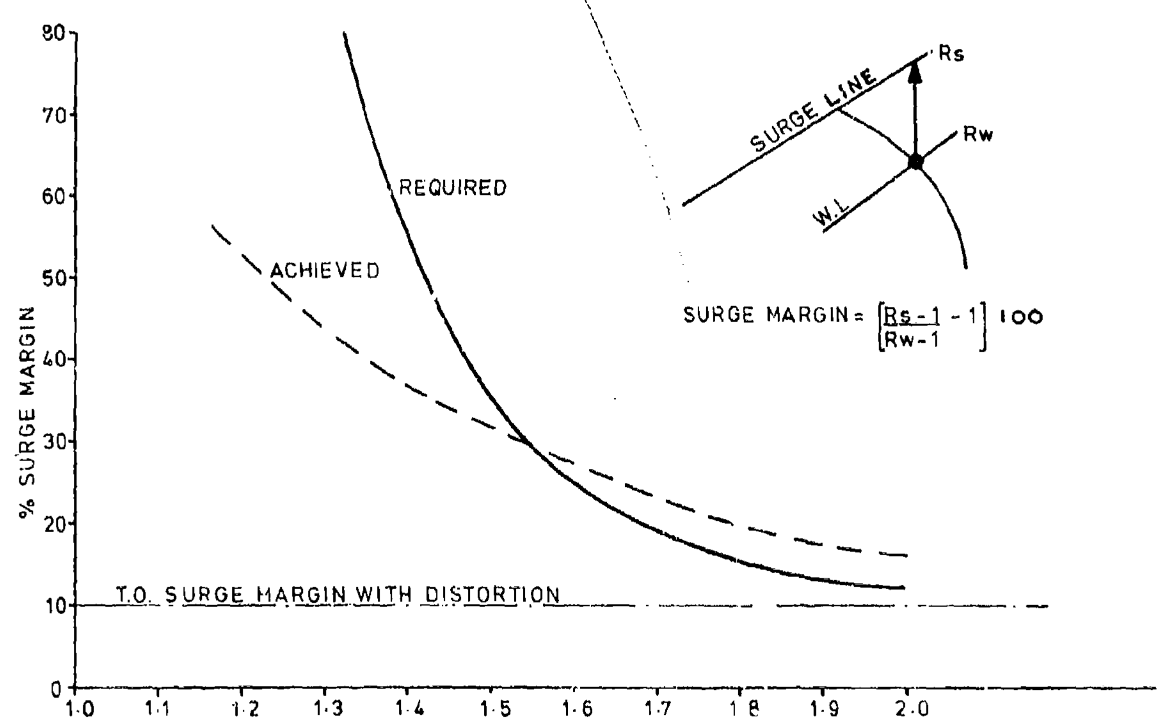


Fig.14 Effect of fan pressure ratio on T.O. surge margin

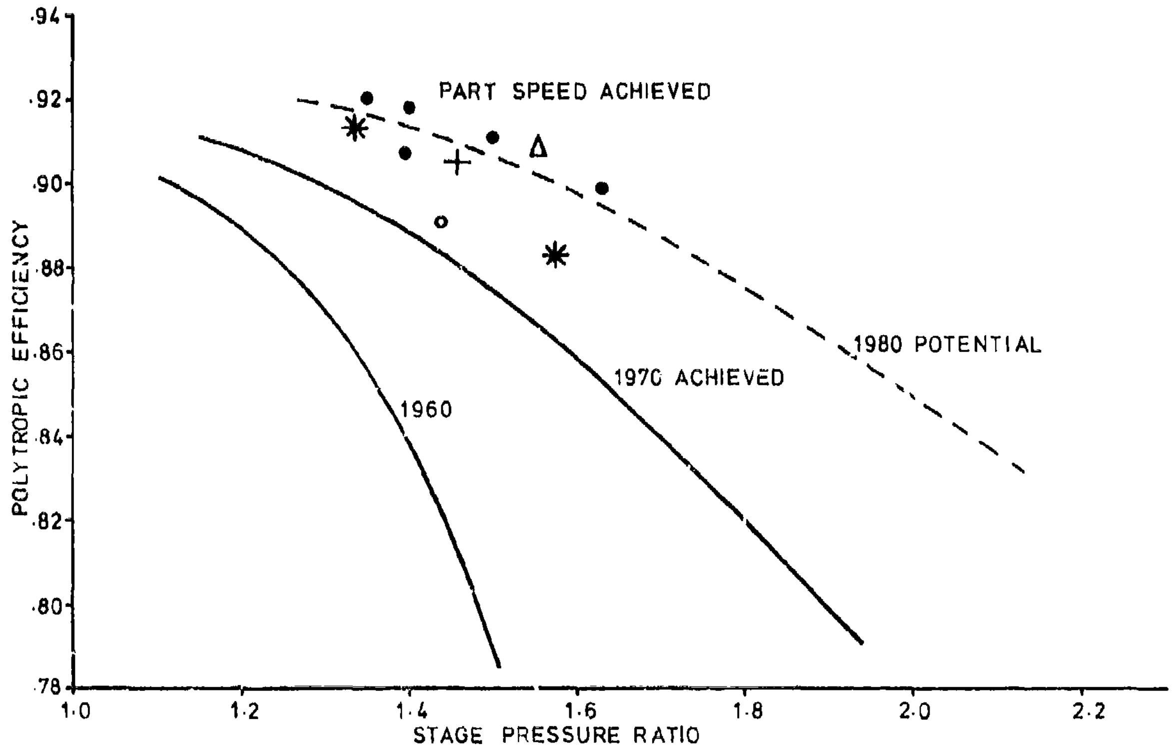


Fig.15A Compressor stage efficiency

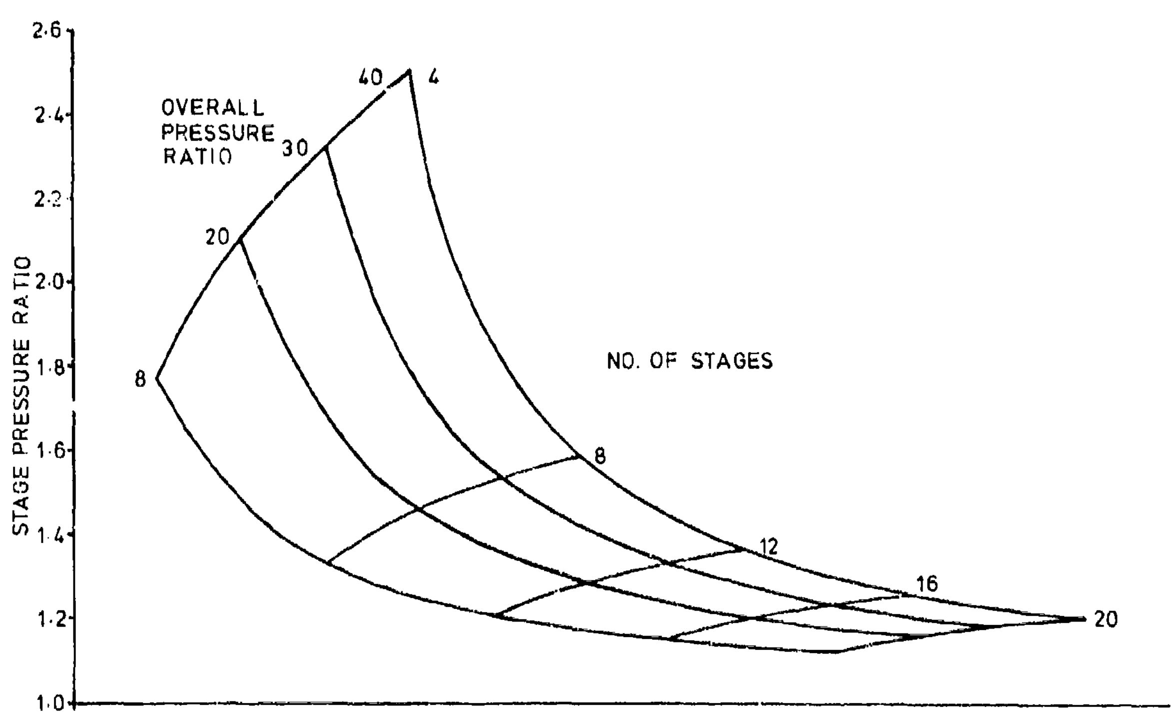


Fig.15B Multistage performance potential

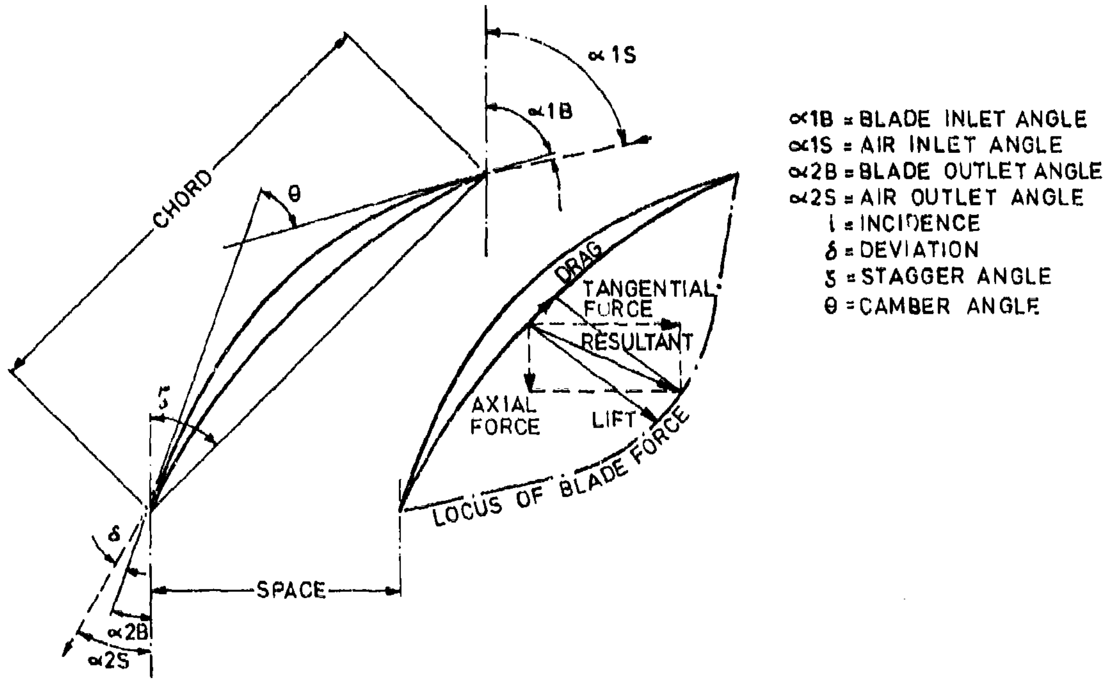


Fig.16A Blade flow geometry

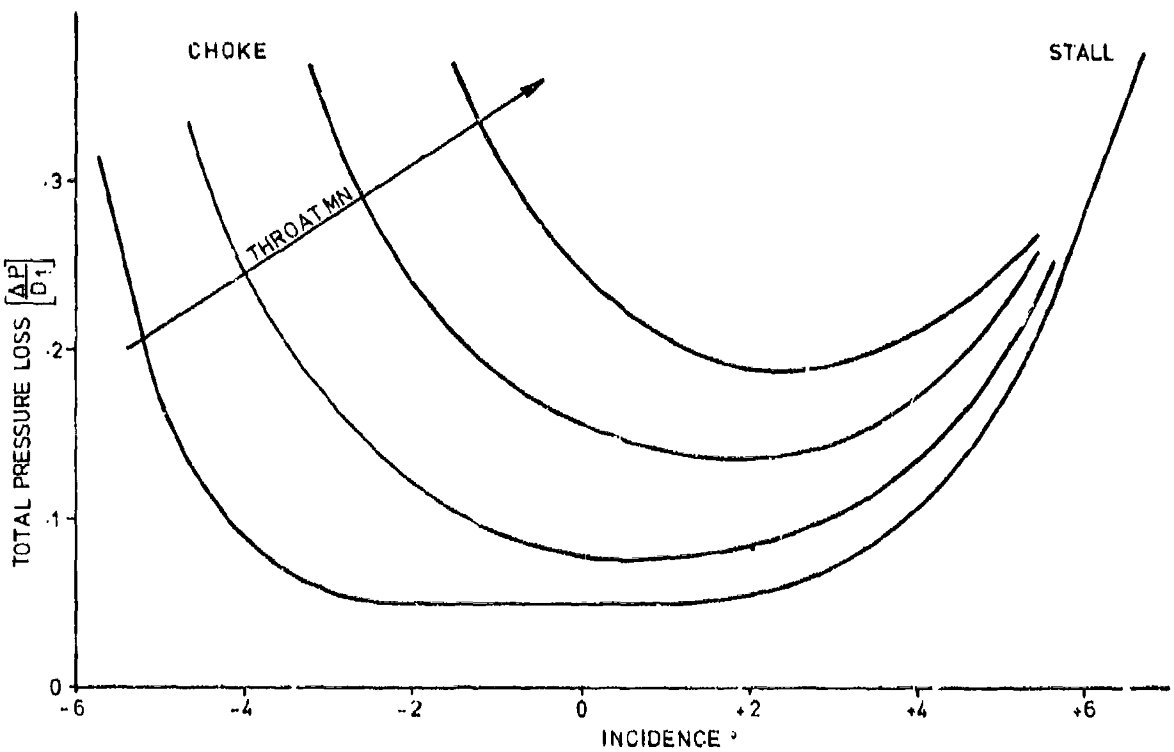


Fig.16B Blade loss characteristics

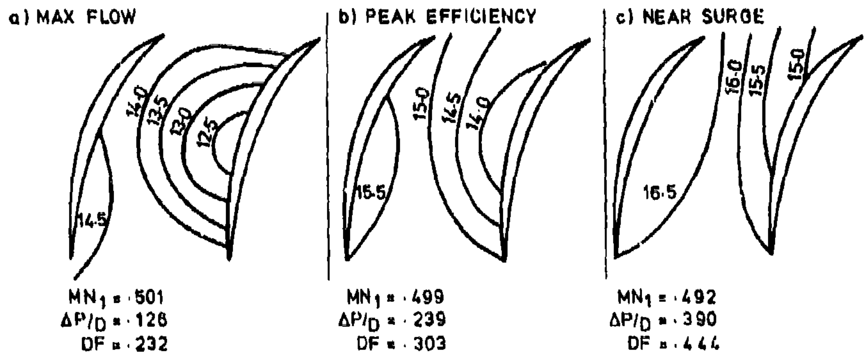


Fig.17 Cascade static pressure field

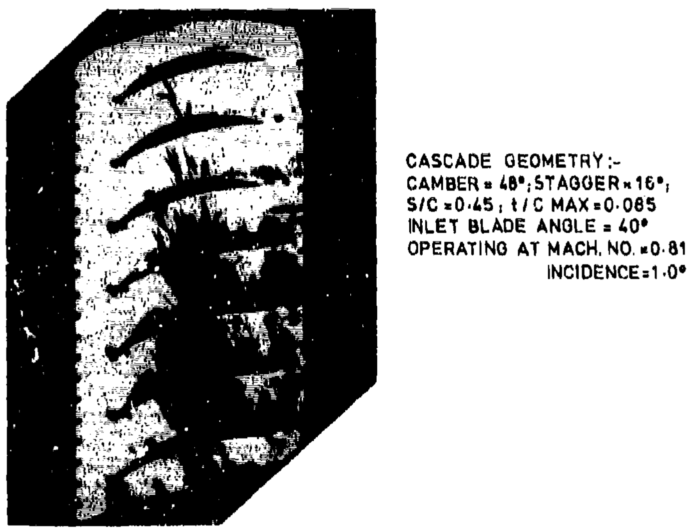


Fig.18 High Mach No. cascade

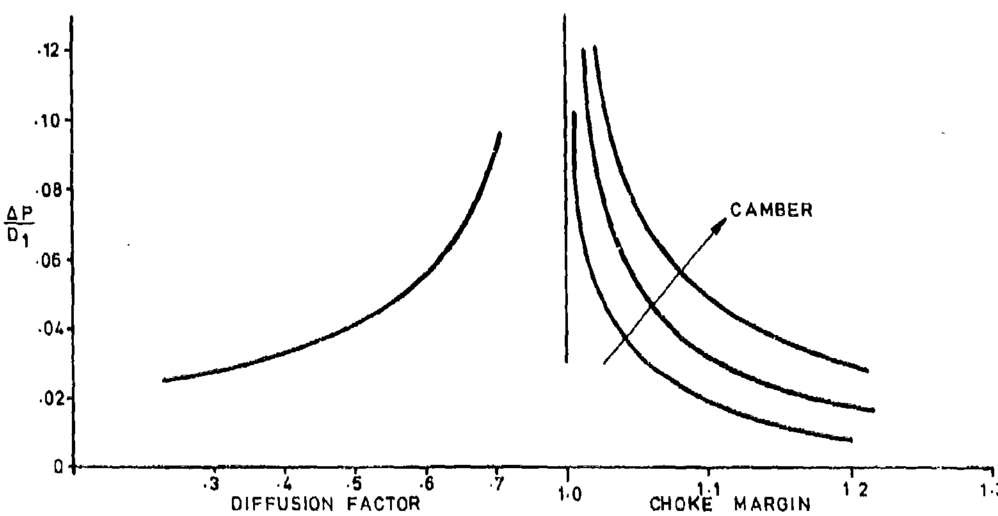


Fig.19 Cascade total pressure loss prediction

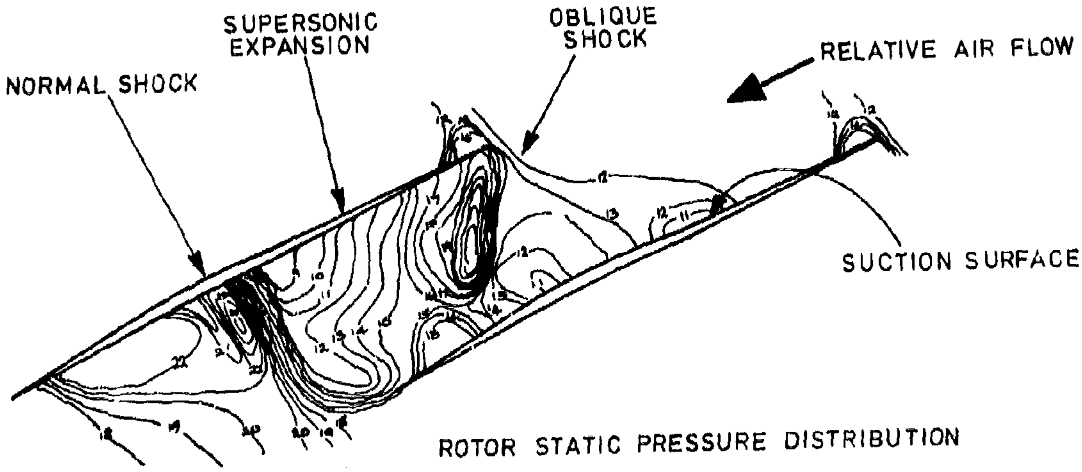


Fig.20 Transonic cascade pressure field

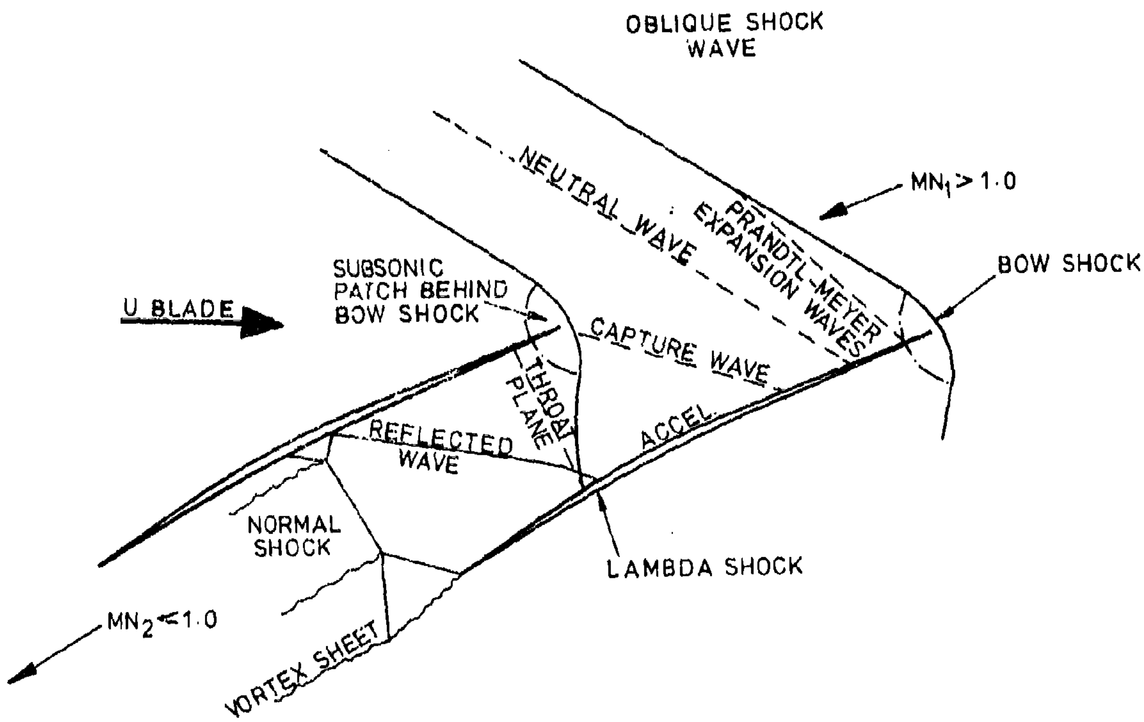


Fig.21 Transonic flow shock model

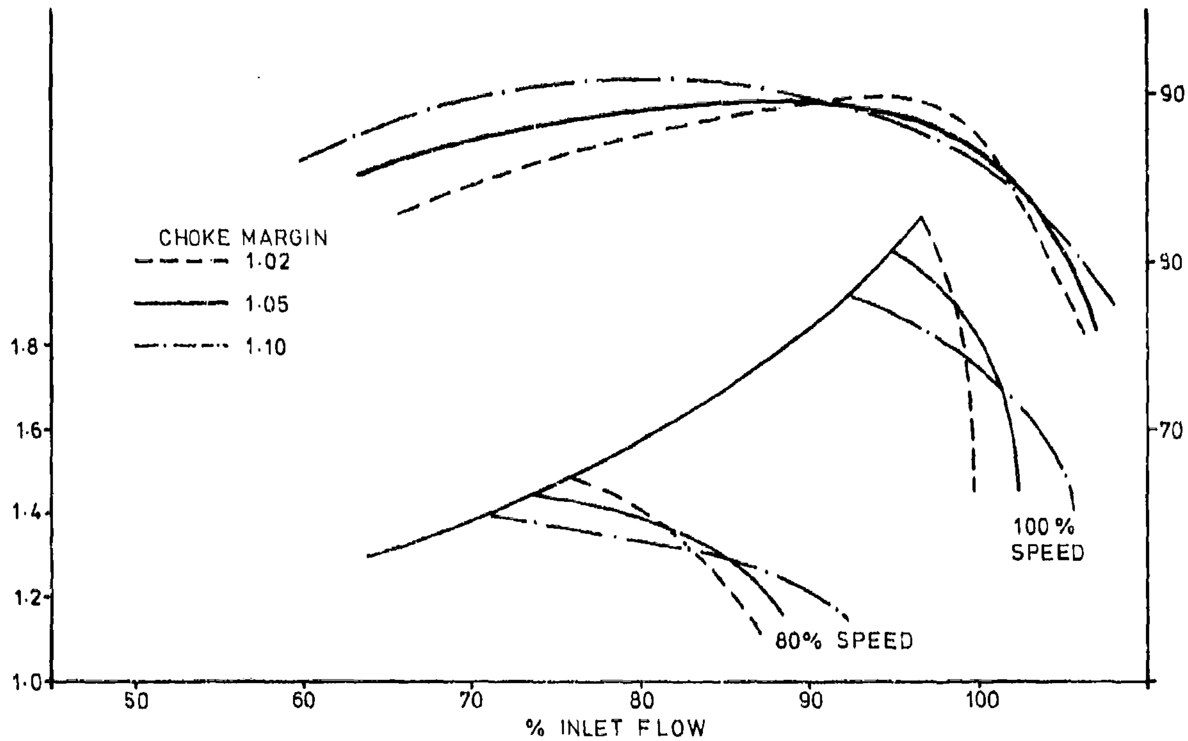


Fig.22 Effect of choke margin on fan characteristics

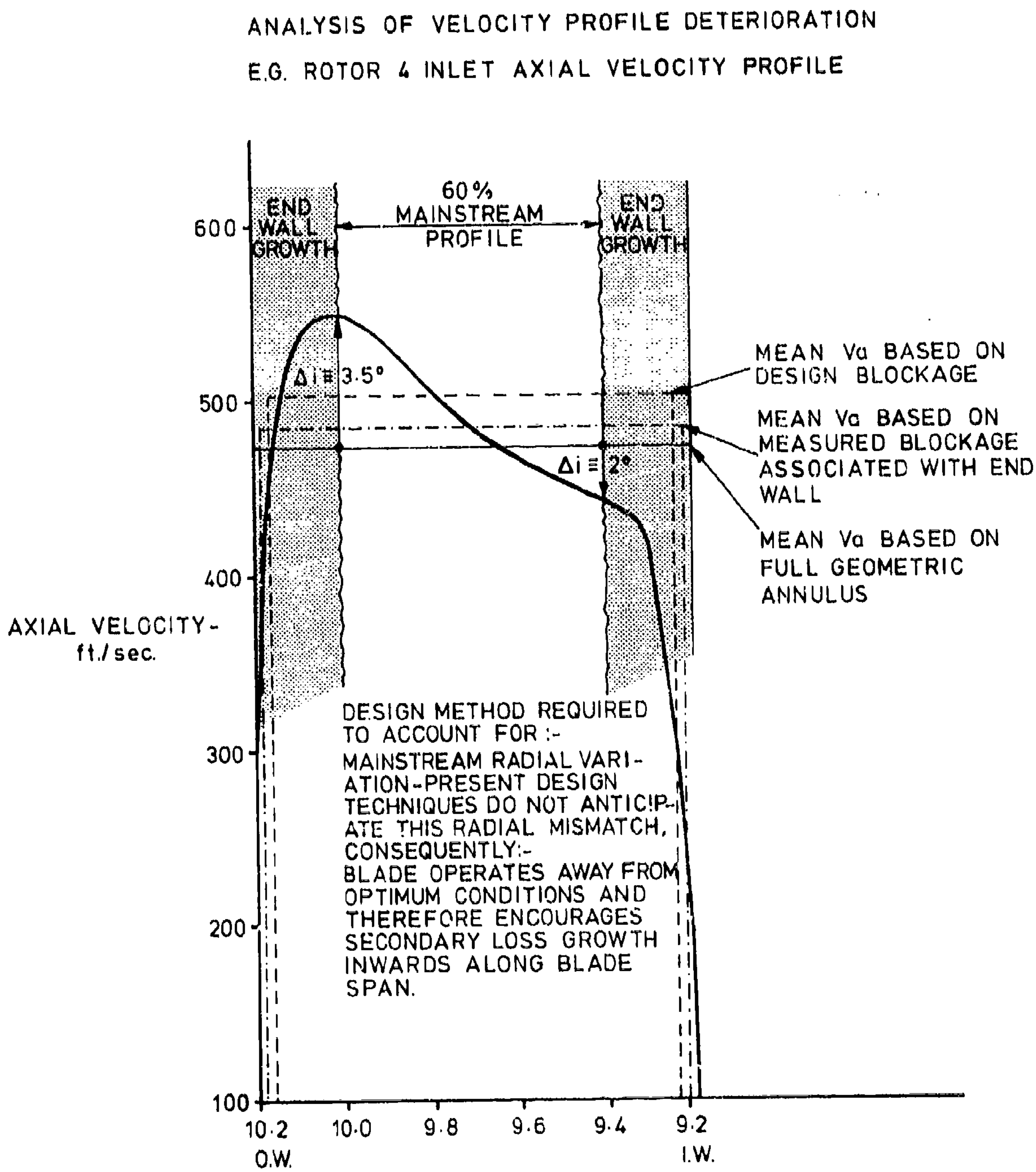
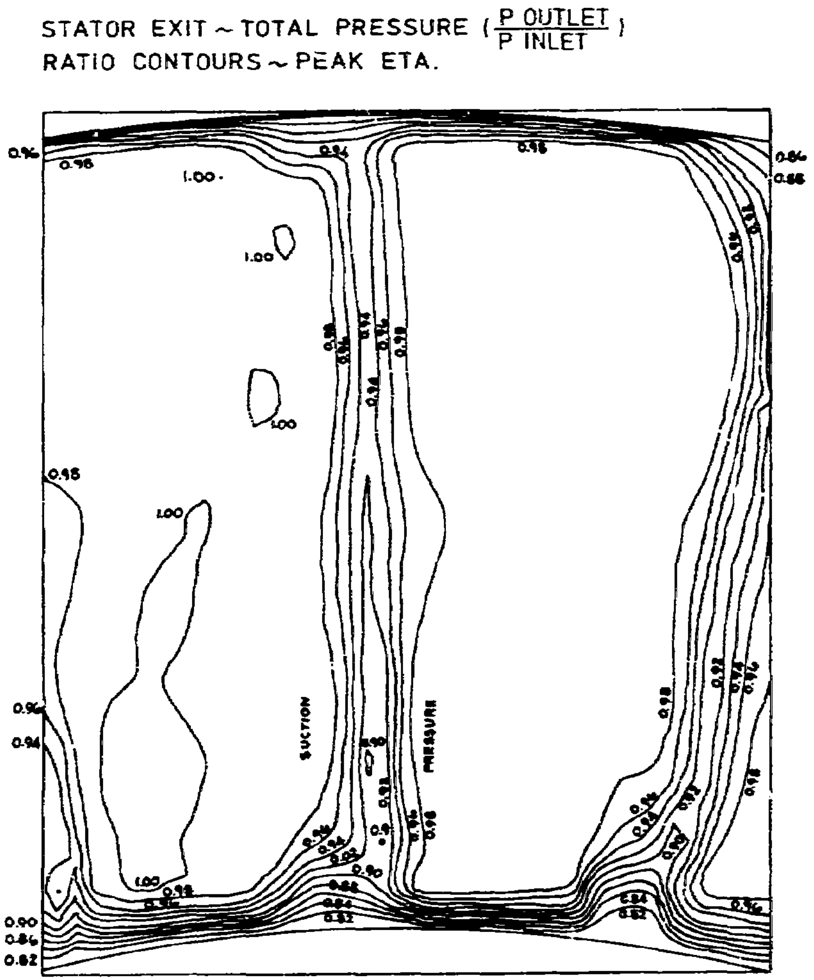


Fig.23 Velocity profile deterioration in multistage



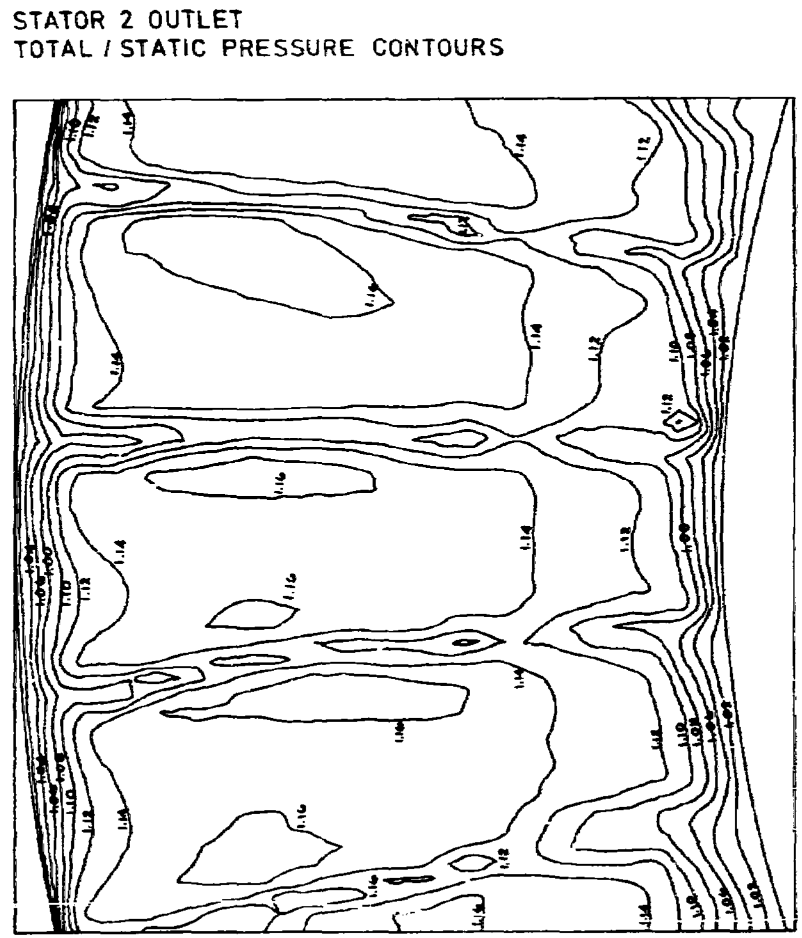


Fig.24 Stage exit pressure fields

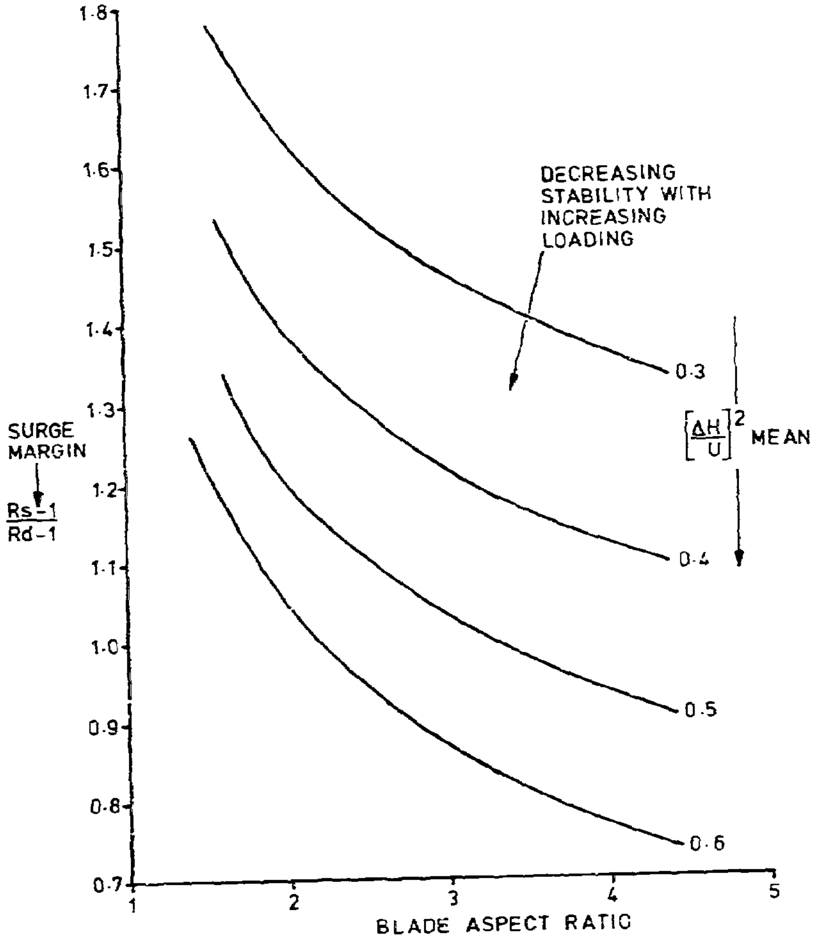


Fig 25 Effect of aspect ratio/loading on surge margin

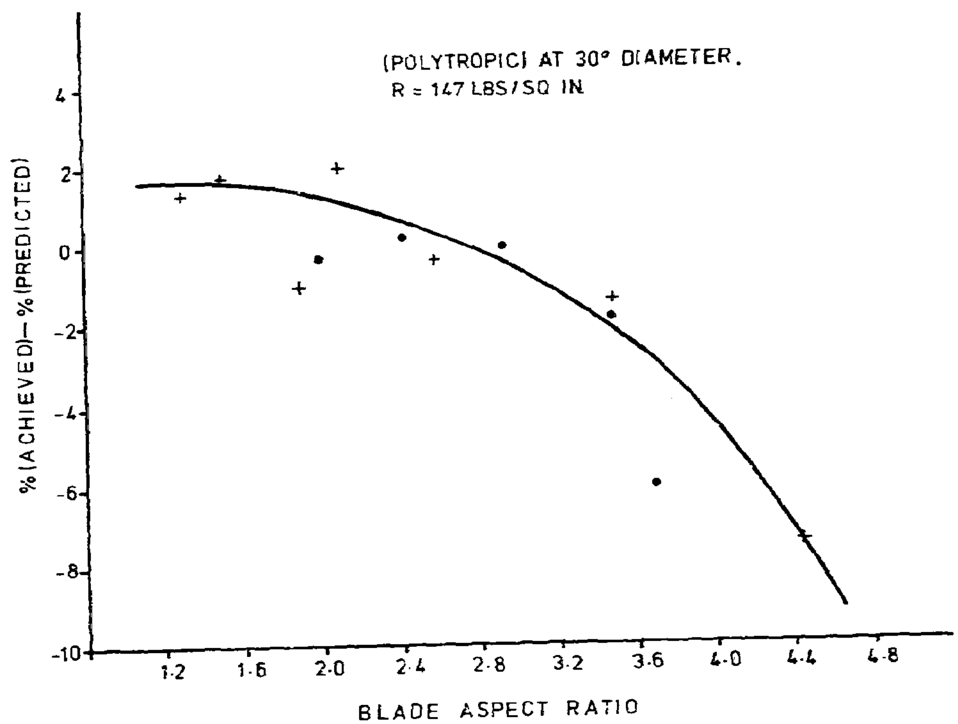


Fig.26 Effect of aspect ratio on efficiency

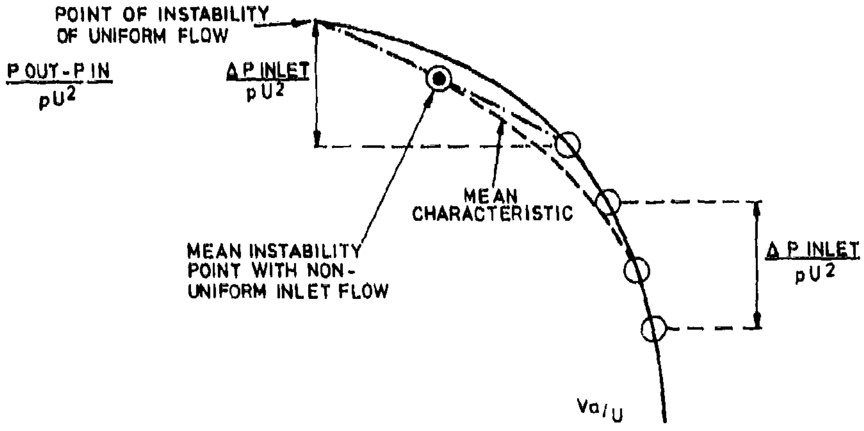


Fig.27A Stage operation with non uniform inlet pressure

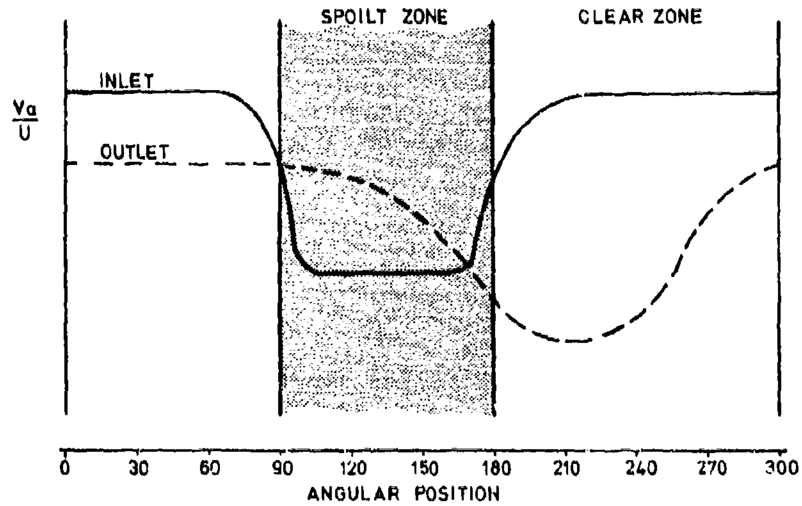


Fig.27B Transient response to distortion

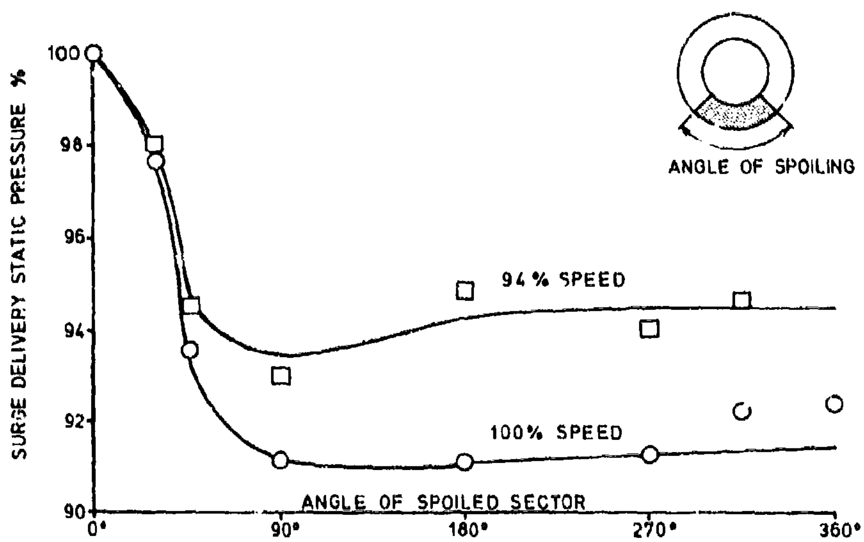
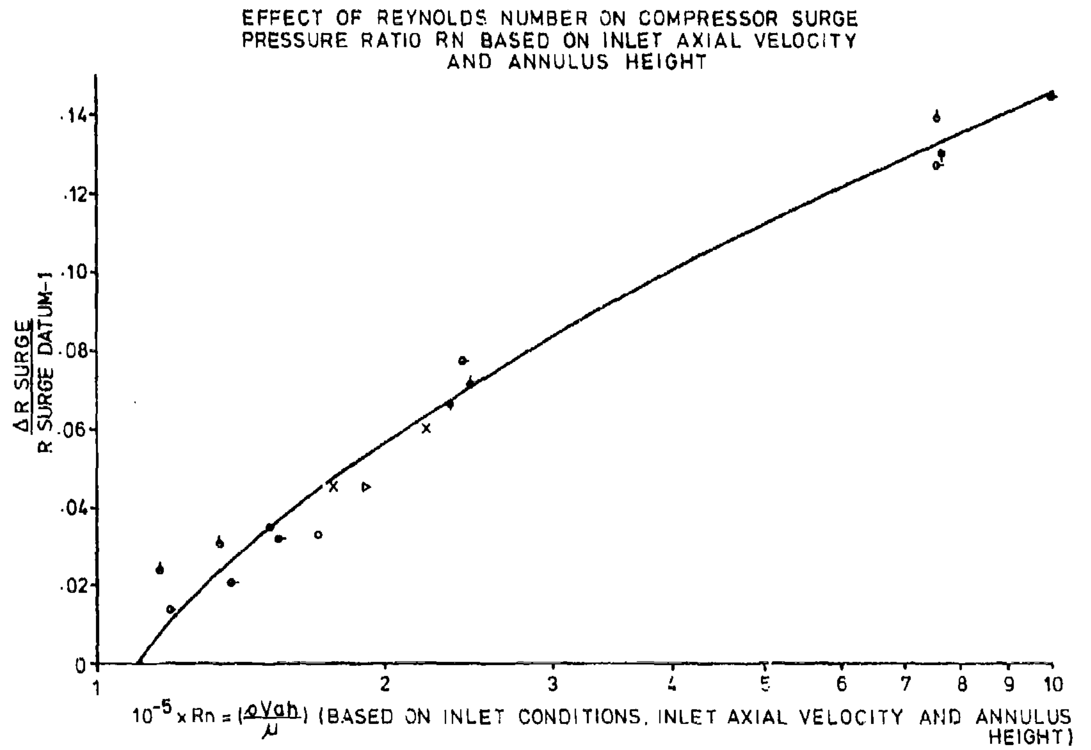


Fig.28 Effect of varying circumferential angle of spoiling



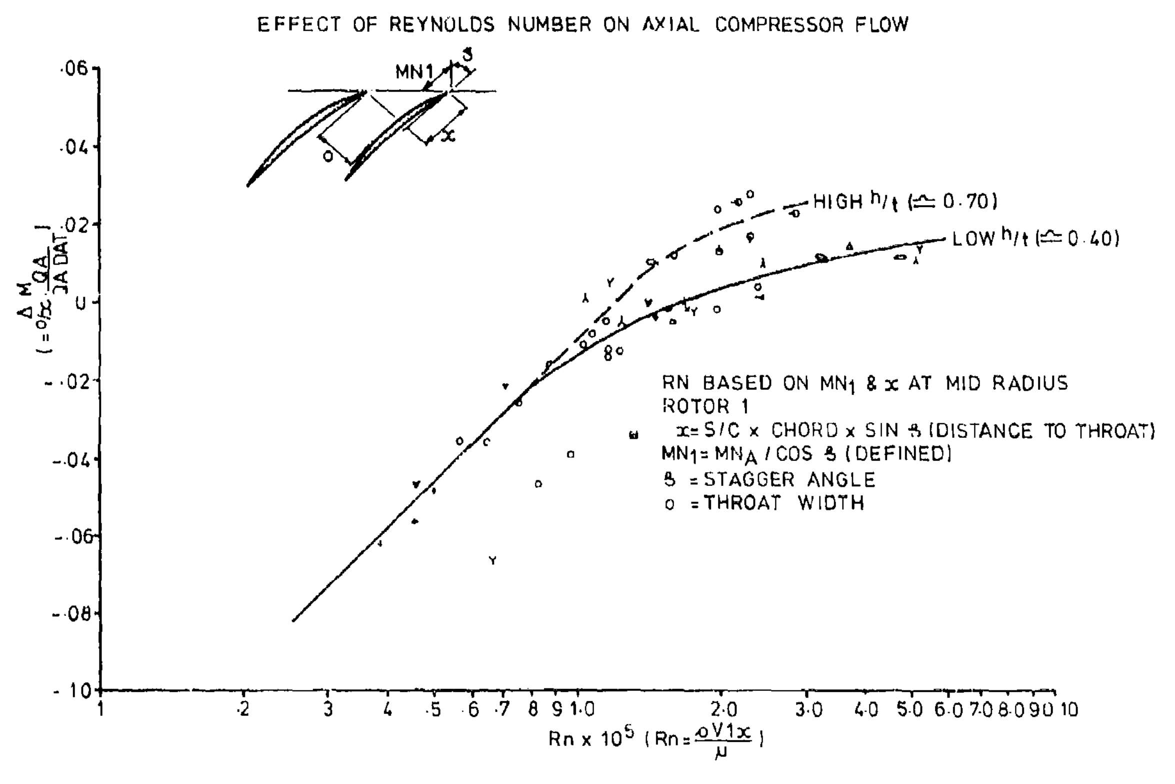


Fig.30 Effect of Reynolds No. on performance

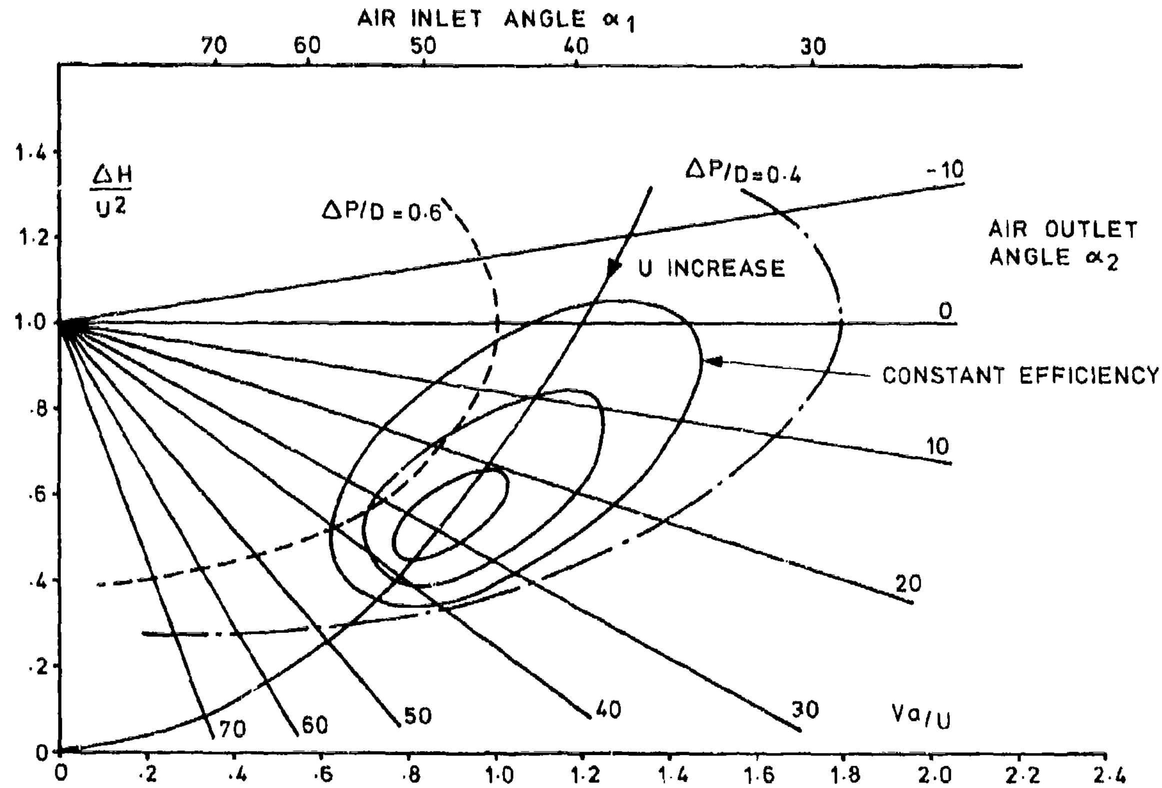


Fig.31 Stage loading for zero α0 design

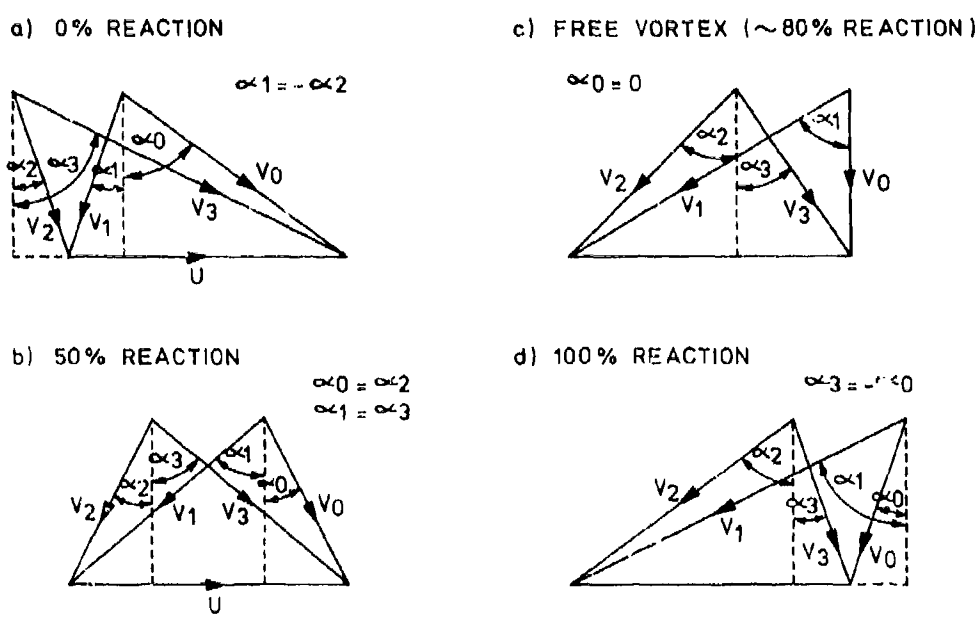
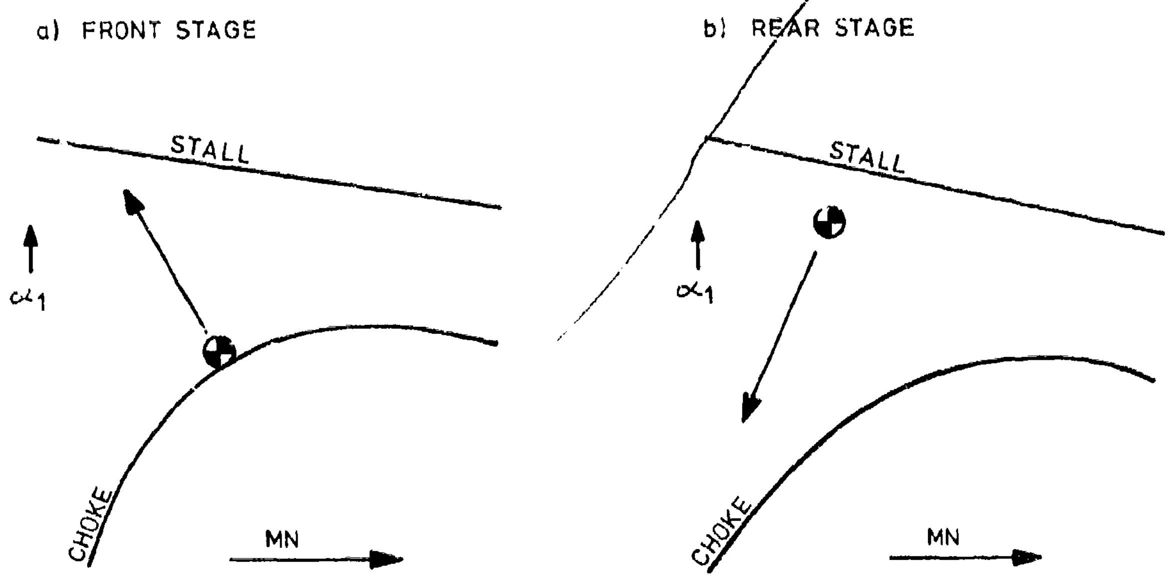


Fig.32 Velocity triangles for different reactions



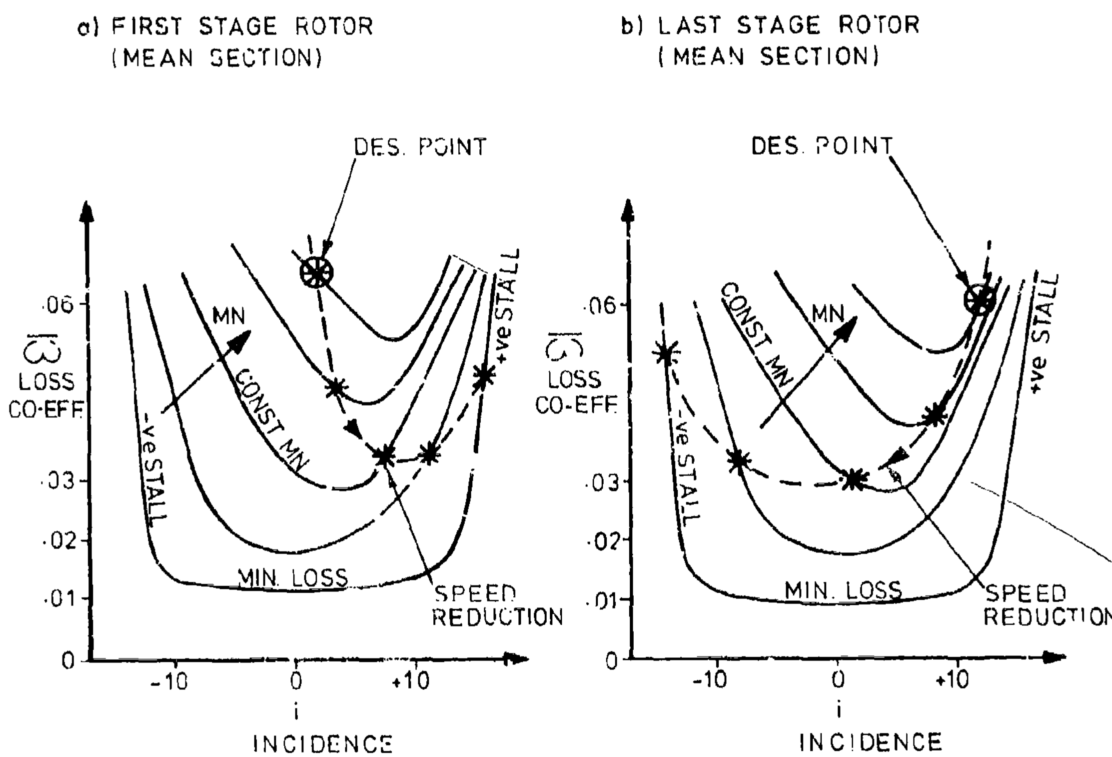


Fig.33 Performance of first/last compressor stages

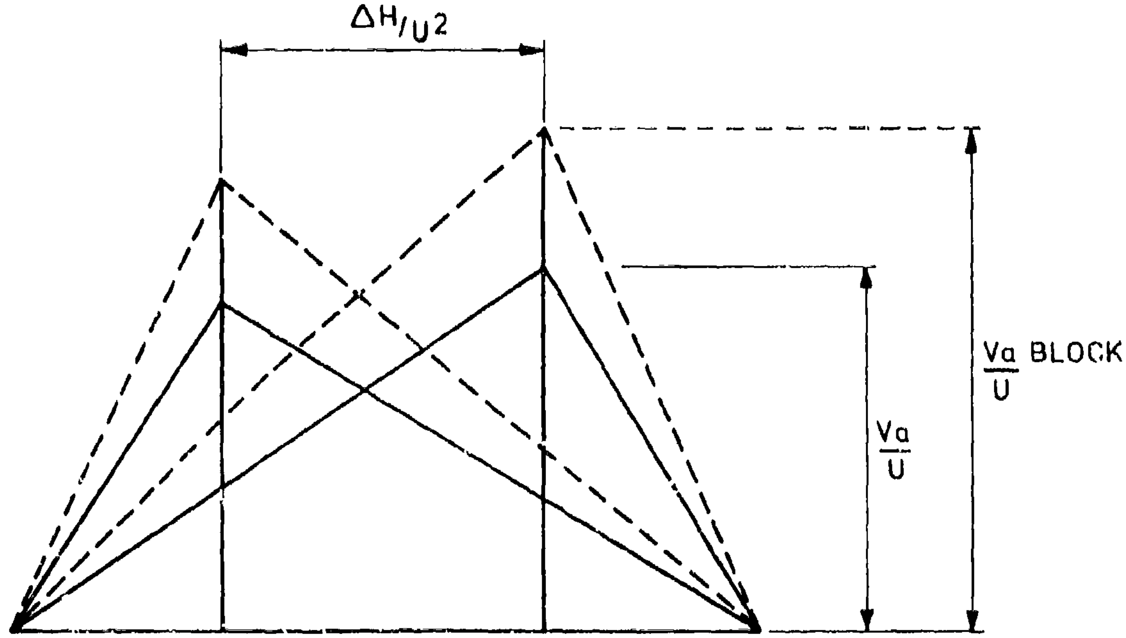


Fig.34A Blockage design vectors

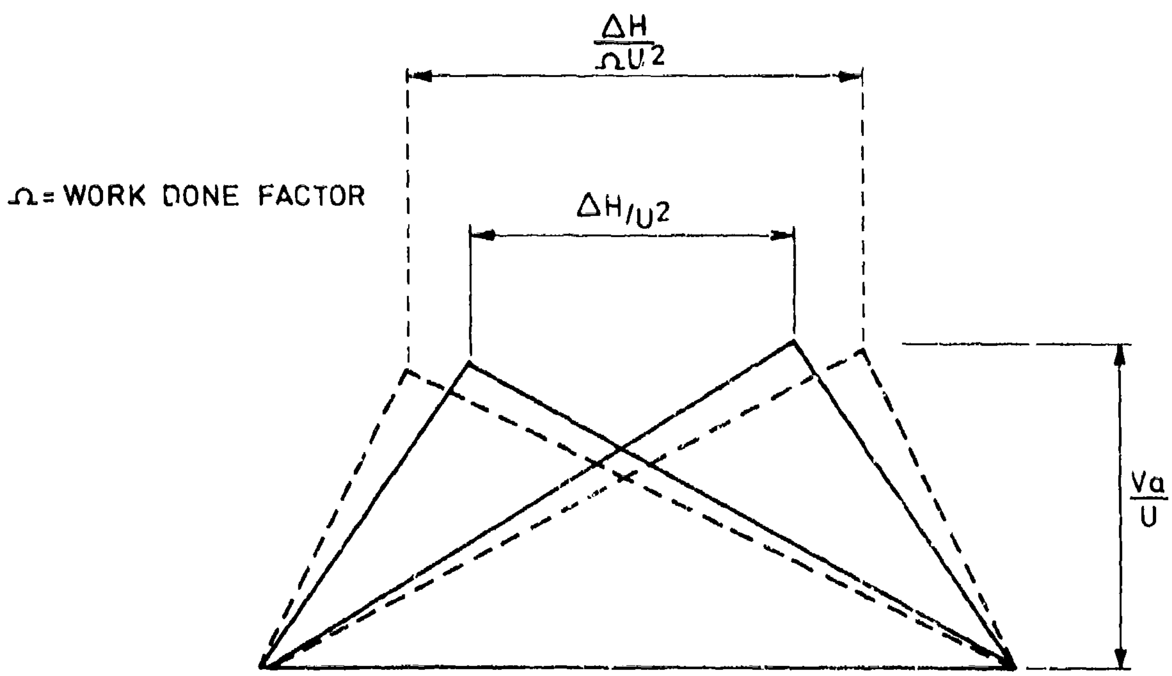


Fig.34B Work done factor design vectors

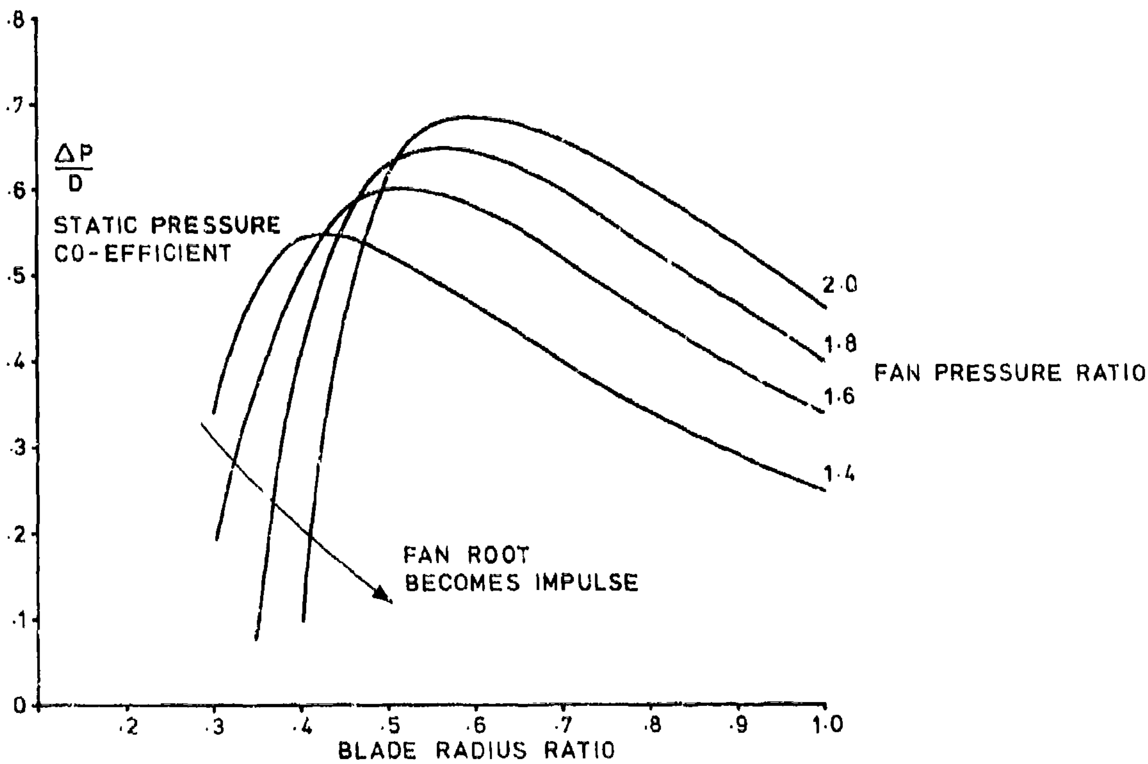


Fig.35A Transonic fan loading

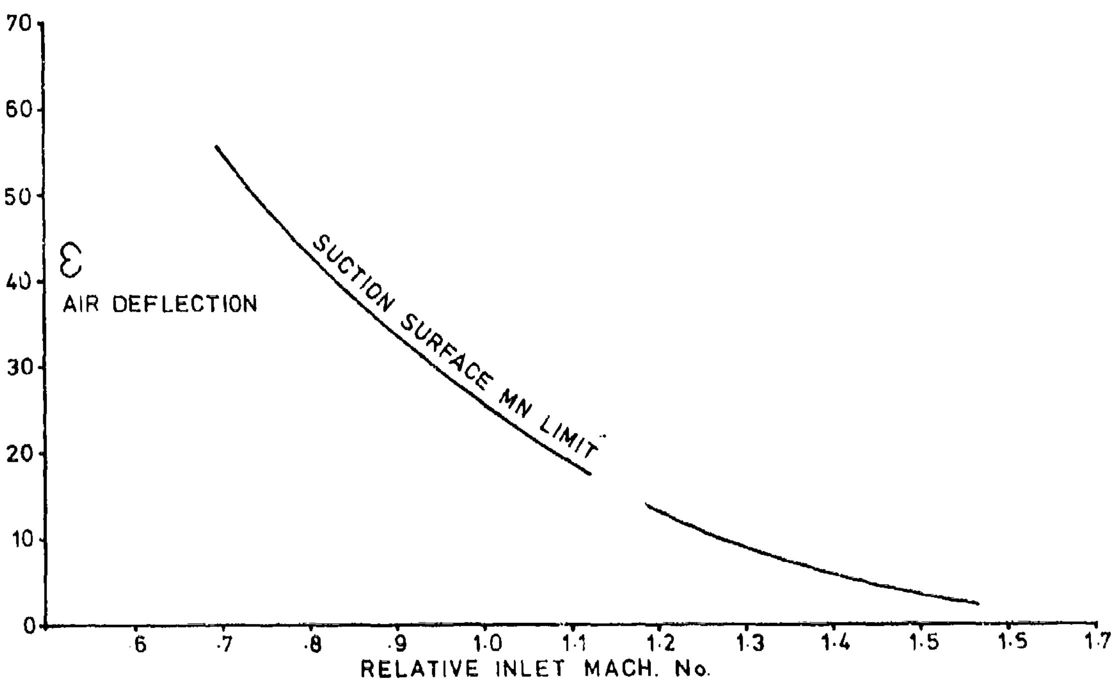


Fig.35B Transonic fan deflection load

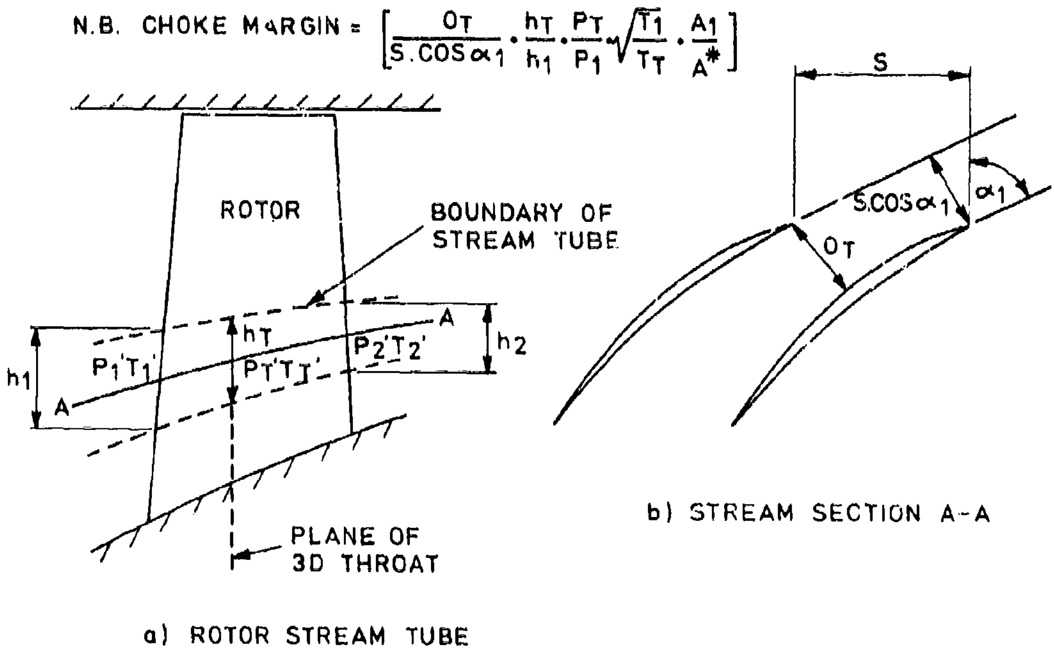


Fig.36 Calculation of choke margin

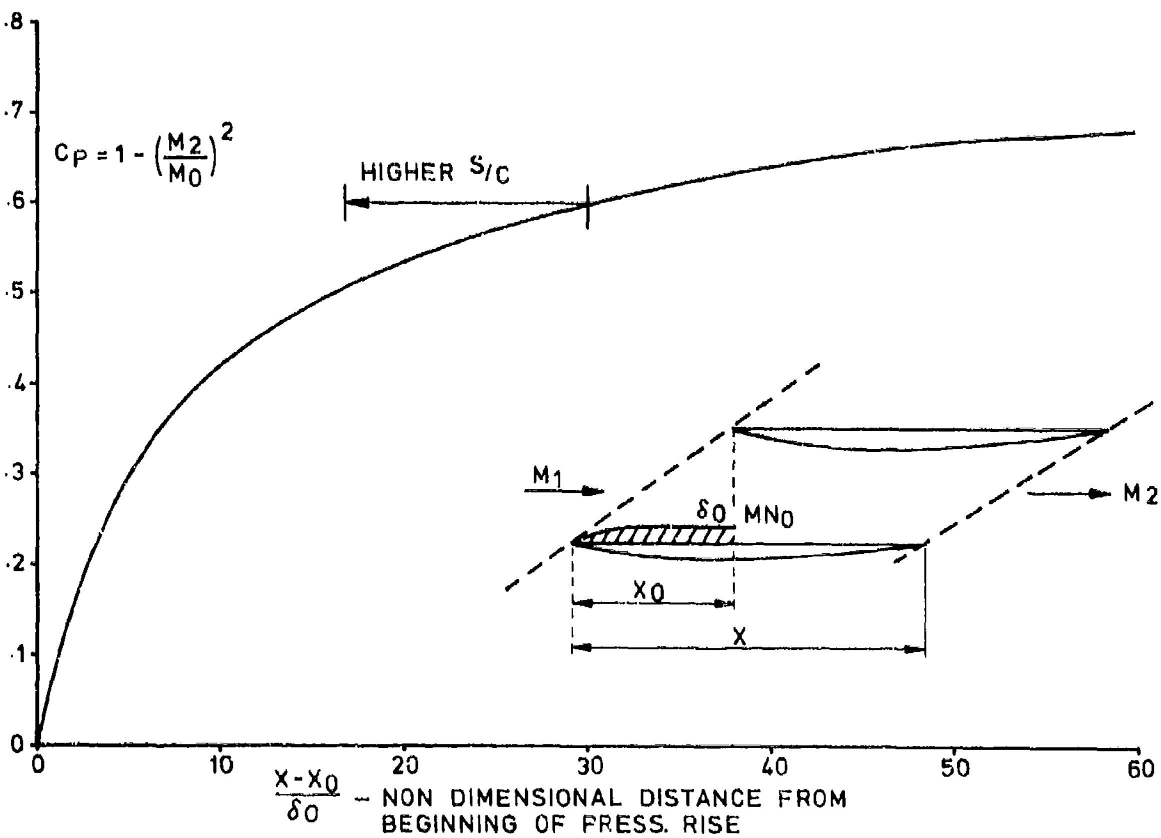


Fig.37 Transonic blade solidity

FLOW FIELD AND PERFORMANCE MAP

COMPUTATION FOR AXIAL-FLOW COMPRESSORS AND TURBINES

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The title given the present Lecture is sufficiently broad so that it is well, before plunging into details, to define the subject matter more precisely. It should be stated at the outset that we intend to concentrate almost wholly on axisymmetric meridional plane computing procedures such as those represented by Refs. 1 and 2. Despite the words of the title, furthermore, attention will chiefly be directed at the axial compressor.

The emphasis on two-dimensional computing schemes, and upon the compressor rather than on the turbine, is not the result of bias. Currently, rapid progress is being made on the implementation of the quasi-three-dimensional technique which was first rather completely defined by Wu (Ref. 3). Refs. 4 and 5 are representative of matrix solutions of Wu's proposals; Refs. 6 and 7 describe an attempt to recast the Wu system into a streamline curvature mold.

Computing techniques whose objective is to define the detailed flow field within a blade row, however, do not yet lend themselves readily to performance prediction. The more complete flow field definition which can be obtained is not so complete that losses can be calculated; it is, furthermore, considerably more expedient to apply empirically determined loss data with a two-dimensional, axisymmetric system.

At the present time, any axial compressor or turbine performance prediction scheme must rely upon the availability of cascade loss and turning data. Such data exist for the axial compressor in sufficient quantity and detail so that quite realistic performance analysis is possible for a wide range of machines. The fact that a similar situation does not exist with respect to axial turbines is the chief reason that the development of axisymmetric prediction techniques for turbines has lagged. As indicated by Serovy in the previous Lecture, axial turbine performance prediction is still best dealt with by one-dimensional approaches such as that of Ainley and Mathiesen. In the present Lecture we shall discuss - but briefly - the problems associated with axisymmetric performance computation for the axial turbine. The discussion and development of the system will be chiefly in the context of axial compressors.

The classical approach in fluid mechanics usually involves an early algebraic combination of the continuity equation with some form of the momentum equation. Traditionally, the mixture results in a second order differential equation, where the dependent variable is the stream function, Ψ. The resulting governing equation looks very much like those found in electrodynamics; it is sometimes amenable to the same techniques of solution.

The process is elegant and appealing. Unfortunately, it does not lend itself well when the fluid is compressible, and there is great difficulty when the fluid passage geometry deviates greatly from what is easily described by a relatively uncomplicated orthogonal coordinate system. The problems associated with arbitrary geometry and compressibility can, of course, be dealt with by mechanized computing techniques. The standard procedure is some form of a finite difference matrix technique. When the computer becomes available, however, there are other ways to proceed, and the streamline curvature approach is one of them.

It is characterized, as distinguished from the classical approach, by the fact that the continuity equation and the momentum equations are kept separate throughout the process. They are independently applied and separately satisfied by successive iteration. It is evident, therefore, that an important consideration with the streamline curvature approach revolves around techniques to make the iterative process as efficient as possible.

The formal part of the presentation follows quite closely that of Smith (Ref. 1) and of Novak (Ref. 2). What will be discussed differs from the material of these two references not in substance, but rather in emphasis. First, we shall devote some attention to solution techniques themselves, and to the means available to speed up convergence. Closely related to this is the problem - one of the more difficult - associated with solutions when the flow in the annulus approaches the maximum possible flow.

One of the more interesting aspects of the streamline curvature approach as applied to axial turbomachinery is the comparative ease with which it can be made into an effective tool for performance prediction. It must, of course, be combined with some formulation of empirical data, i.e., cascade loss and turning data. We will discuss one particular version of a cascade subroutine and, finally, finish with the presentation of a number of cases where computed and experimental results can be compared.

One final overall acknowledgement is in order. The author has, for many years, worked closely with R. M. Hearsey, both on the quasi-three-dimensional system of Refs. 6 and 7, and on the axisymmetric technique which is the model for much of the following discussion. The particular computing program which generated the examples discussed later is wholly the work of Hearsey.

THE MOMENTUM EQUATION - THE DESIGN PROBLEM

With respect to a cylindrical coordinate system rotating with a constant angular velocity, $\bar{\omega}$, the momentum equation is written as:

$$-\frac{1}{\rho}\nabla p = \frac{dW}{dt} - \bar{\omega}^2 r + 2(\bar{\omega} \times W) \tag{1}$$

where W is the velocity relative to the coordinate system, ρ is the density and p is the static pressure. Also,

$$\frac{dW}{dt} = \frac{\partial W}{\partial t} + W\frac{dW}{dS} \tag{2}$$

We observe at this point that we will consistently deal with the momentum-equations in the steady-state form, whether we deal with flow on a rotor or on a stator. For a truly axisymmetric system no problem is thereby raised; this is distincly not the case if any degree of circumferential nonuniformity exists. It is beyond the scope of the present effort, however, to deal with nonsteady problems.

We call the three components of W to be V_r, W_θ and V_z (since $V_r = W_r$ and $V_z = W_z$). W_θ is given by:

$$W_\theta = V_\theta - \bar{\omega}r \tag{3}$$

We define, furthermore, a velocity component, V_m, a direction, m, and an angle, ϕ, such that:

$$V_r = V_m \sin\phi$$

$$V_z = V_m \cos\phi \tag{4}$$

We adopt the convention of signifying a differentiation following a particle by d/dS where S is the streamwise direction. Thus,

$$W\frac{d}{dS} = V_r\frac{\partial}{\partial r} + \frac{W_\theta}{r}\frac{\partial}{\partial\theta} + V_z\frac{\partial}{\partial z} \tag{5}$$

Finally, by virtue of Eq. (4):

$$\frac{\partial}{\partial m} = \sin\phi\frac{\partial}{\partial r} + \cos\phi\frac{\partial}{\partial z} \tag{6}$$

Using the definition above, the three component equations of Eq. (1) become:

$$-\frac{1}{\rho}\frac{\partial p}{\partial r} = V_m\frac{\partial V_r}{\partial m} + \frac{W_\theta}{r}\frac{\partial V_r}{\partial\theta} - \frac{V_\theta^2}{r} \tag{7}$$

$$-\frac{1}{\rho}\frac{\partial p}{\partial\theta} = V_m\frac{\partial rV_\theta}{\partial m} + \frac{W_\theta}{r}\frac{\partial rV_\theta}{\partial\theta} \tag{8}$$

$$-\frac{1}{\rho}\frac{\partial p}{\partial z} = V_m\frac{\partial V_z}{\partial m} + \frac{W_\theta}{r}\frac{\partial V_z}{\partial\theta} \tag{9}$$

where Eq. (3) has been used to express Eq. (8) in terms of derivatives of V_θ instead of W_θ.

The present Lecture deals basically with flow field solution techniques which can be reasonably represented by the axisymmetric equations. If the θ-derivative terms are dropped from the equations above, there results:

$$-\frac{1}{\rho}\frac{\partial p}{\partial r} = V_m\frac{\partial V_r}{\partial m} - \frac{V_\theta^2}{r} \tag{10}$$

$$0 = V_m\frac{\partial rV_\theta}{\partial m} \tag{11}$$

$$-\frac{1}{\rho}\frac{\partial p}{\partial z} = V_m\frac{\partial V_z}{\partial m} \tag{12}$$

Eq. (11) expresses the condition that there can be no angular momentum change in the streamline direction - or along the meridional projection of a streamline - in an axisymmetric flow field. The condition is also the one which gives rise to logical devices (i.e., infinite blade number, distributed body forces, etc.) with which to justify the use of pseudo-axisymmetric equations within a blade passage. The resort to such devices is not simply a whim; it results from a very realistic need to avoid, if possible, the necessity for dealing with three dimensions.

We introduce Eqs. (4) into (10) and (12) and perform the indicated differentiation. The result is:

$$-\frac{1}{\rho}\frac{\partial p}{\partial r} = \frac{V_m^2\cos\phi}{r_m} + V_m\sin\phi\frac{\partial V_m}{\partial m} - \frac{V_\theta^2}{r} \tag{13}$$

$$-\frac{1}{\rho}\frac{\partial p}{\partial z} = -\frac{V_m^2\sin\phi}{r_m} + V_m\cos\phi\frac{\partial V_m}{\partial m} \tag{14}$$

where

$$\frac{\partial \phi}{\partial m} = + \frac{1}{r_m} \qquad (15)$$

and r_m is the radius of curvature of the meridional projection of the streamline.

Fig. 1 and Fig. 2 illustrate the terms which have been defined in the foregoing. Fig. 2 (a meridional plane picture of a passage) also shows a computing station, and a direction along it denoted as $\bar{q}$. This is the direction which has been labeled by Katsanis (Ref. 8) as the quasi-orthogonal direction.

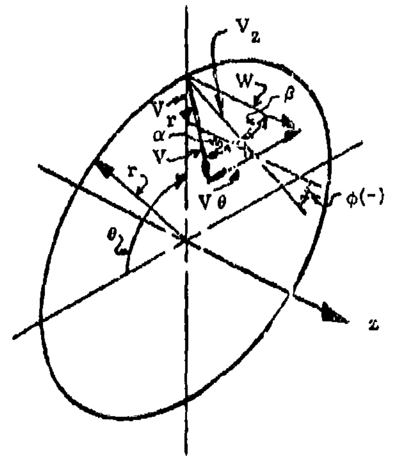

Figure 1

We write:

$$\frac{\partial}{\partial \bar{q}} = \bar{A}\frac{\partial}{\partial r} + \bar{C}\frac{\partial}{\partial z} \qquad (16)$$

or specifically,

$$-\frac{1}{\rho}\frac{\partial p}{\partial \bar{q}} = -\frac{\bar{A}}{\rho}\frac{\partial p}{\partial r} - \frac{\bar{C}}{\rho}\frac{\partial p}{\partial z}$$

where $\bar{A}$ and $\bar{C}$ are direction cosines which the $\bar{q}$-direction makes with the r- and z- axes.

The angle γ is defined by Fig. 2; it is clear that:

$$\bar{A} = \cos\gamma$$
$$\bar{C} = \sin\gamma$$

The above, with Eqs. (10) and (12), results in:

$$\frac{1}{\rho}\frac{\partial p}{\partial \bar{q}} = \frac{V_\theta^{\,2}\cos\gamma}{r} - \frac{V_m^{\,2}\cos(\gamma + \phi)}{r_m} - \sin(\gamma + \phi)V_m\frac{\partial V_m}{\partial m} \qquad (17)$$

It is evident by noting the manner in which the angles γ and ϕ were defined (Fig. 2) that Eq. (17) becomes greatly simplified if the quasi-orthogonal is truly orthogonal to the meridional projection of the streamline. In this event $\sin(\gamma + \phi) = 0$ and $\cos(\gamma + \phi) = 1$. The m-derivative of V_m disappears from the equation.

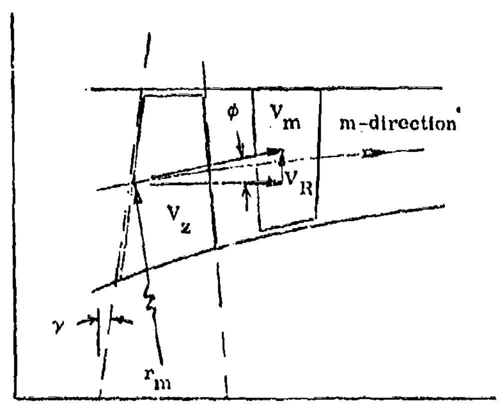

Figure 2

In the context of the steps in the development which follow, it would be happy not to be required to evaluate $\partial V_m/\partial m$. The attendant difficulty of working with orthogonals whose location is not known a priori becomes untenable, however, especially if the problem deals with a multistage compressor or turbine where the axisymmetric solution really involves stations placed in the gaps between blades.

In this connection it is well to point out that the computational problem is not made significantly worse (for a computer) if the γ-angle varies with the $\bar{q}$-distance. If the computer program is set up such that any $\bar{q}$-station is defined by a series of arbitrary r-, z-coordinates along its length, so that, for example the exact leading and trailing edges of each blade become the computing stations, the maximum flexibility for the designer can be attained. When the $\bar{q}$-locus is specified by coordinates in an input data deck, the computer will obtain its own values of γ for use in Eq. (17).

We next operate on the $\partial V_m/\partial m$-term, i.e., the acceleration term arising from a rate of change of meridional velocity in the meridional direction. Although we are dealing in the present Lecture with axisymmetric systems (i.e., a blade-free space), it is expedient to recognize the possibility of a blockage. We consider it as a distributed blockage to avoid a conflict with the condition of axial symmetry. The reasons for the introduction of a blockage-term will become evident below.

The blockage is denoted by $\bar{\lambda}$; $\lambda = (1 - \bar{\lambda})$ is then the open area. For our axisymmetric model with distributed blockage, the continuity equation is written as:

$$\frac{\partial(r\rho\lambda V_r)}{\partial r} + \frac{\partial(r\rho\lambda V_z)}{\partial z} = 0 \qquad (18)$$

The choice has been made to use the m-direction and the $\bar{q}$-direction as independent variables. Eqs. (6) and (16) imply that:

$$\frac{\partial}{\partial r} = -\frac{1}{\cos(\gamma + \phi)}\left[\sin\gamma\frac{\partial}{\partial m} - \cos\phi\frac{\partial}{\partial \bar{q}}\right] \qquad (19)$$

$$\frac{\partial}{\partial z} = \frac{1}{\cos(\gamma + \phi)} \left[\cos\gamma \frac{\partial}{\partial m} - \sin\phi \frac{\partial}{\partial \bar{q}}\right] \tag{20}$$

Using these two operators on Eq. 18, and resorting to the definitions of V_m and r_m as before, one obtains:

$$V_m^2 \left[\frac{\partial \ell n r\rho\lambda}{\partial m} - \frac{\tan(\gamma + \phi)}{r_m} + \sec^2(\gamma + \phi) \frac{\partial \phi}{\partial \bar{q}}\right] + V_m \frac{\partial V_m}{\partial m} = 0 \tag{21}$$

We use the pressure-density relationship and the equation of state to transform the first term of Eq. (21). Thus,

$$\frac{p}{\rho} = \frac{\gamma - 1}{\gamma} h$$

$$k\rho^{\gamma - 1} = \frac{\gamma - 1}{\gamma} h$$

and finally

$$r\rho\lambda = r\lambda \left[\frac{1}{k} \left(\frac{\gamma - 1}{\gamma}\right) h\right]^{\frac{1}{\gamma - 1}} \tag{22}$$

where h is the static enthalpy, k is a constant and γ is the isentropic exponent (easily distinguished from the γ- angle by its use).

We note that

$$h = H - \frac{1}{2} V^2 \tag{23}$$

where H is the stagnation enthalpy of the fluid, and

$$I = H - \bar{\omega}(rV_\theta) = h + \frac{1}{2} W^2 - \frac{1}{2} (\bar{\omega} r)^2 \tag{24}$$

where I is a quantity which has been called the rothalpy and, by some, the relative stagnation enthalpy. It is a quantity which is a constant along a streamline in a rotating system, as is H a constant in a stationary system (in the absence of heat transfer). For an axisymmetric system the two quantities remain constant on stream surfaces and, therefore, in the m- direction.

From Eq. (24), we have

$$h = I - \frac{1}{2} (V_m)^2 - \frac{1}{2} (W_\theta)^2 + \frac{1}{2} (\bar{\omega} r)^2 \tag{25}$$

and

$$\frac{\partial h}{\partial m} = -V_m \frac{\partial V_m}{\partial m} - \left[V_\theta^2 - 2(\bar{\omega} r)V_\theta\right] \frac{\sin\phi}{r} \tag{26}$$

where we have used Eq. (3) and Eq. (11).

The logarithmic derivative of Eq. (22) is

$$\frac{\partial(\ell n\rho r\lambda)}{\partial m} = \frac{\sin\phi}{r} + \frac{1}{\lambda} \frac{\partial \lambda}{\partial m} + \frac{1}{(\gamma - 1)h} \frac{\partial h}{\partial m}$$

Using Eq. (26), and noting that $(\gamma - 1)h$ is the square of the sound velocity, a, this becomes:

$$\frac{\partial(\ell n\rho r\lambda)}{\partial m} = \frac{\sin\phi}{r} + \frac{1}{\lambda} \frac{\partial \lambda}{\partial m} - \frac{1}{a^2} \left\{ V_m \frac{\partial V_m}{\partial m} + \left[2(\bar{\omega} r)V_\theta - V_\theta^2\right] \frac{\sin\phi}{r} \right\} \tag{27}$$

Finally, combining Eq. (21) and Eq. (27), one obtains

$$V_m \frac{\partial V_m}{\partial m} = \frac{V_m^2}{1 - Mm^2} \left\{ \frac{\tan(\gamma + \phi)}{r_m} - \sec^2(\gamma + \phi) \frac{\partial \phi}{\partial \bar{q}} - \frac{1}{\lambda} \frac{\partial \lambda}{\partial m} - \left[1 - M_\theta^2 + M_\theta \left(\frac{\bar{\omega} r}{a}\right)^2\right] \frac{\sin\phi}{r} \right\} \tag{28}$$

where

$$Mm = V_m/a$$

$$M_\theta = V_\theta/a$$

We return at this point to the momentum equation, Eq. (17). For computational purposes, it is

expedient to eliminate the static pressure gradient term. The First and Second Laws of Thermodynamics yield

$$\frac{1}{\rho}\,dp = dh - td\bar{S} \tag{29}$$

where t is the static temperature and $\bar{S}$ is the entropy. We may write, therefore,

$$\frac{1}{\rho}\frac{\partial p}{\partial \bar{q}} = \frac{\partial h}{\partial \bar{q}} - t\,\frac{\partial \bar{S}}{\partial \bar{q}} = \frac{\partial h}{\partial \bar{q}} - h\,\frac{\partial(\bar{S}/c_p)}{\partial \bar{q}} \tag{30}$$

We use Eq. (25) to obtain the first term and write

$$\frac{1}{\rho}\frac{\partial p}{\partial \bar{q}} = \frac{\partial I}{\partial \bar{q}} - h\,\frac{\partial(\bar{S}/c_p)}{\partial \bar{q}} - \frac{1}{2}\left(\frac{\partial V_m{}^2}{\partial \bar{q}}\right) + \frac{W_\theta}{r}\,\frac{\partial rV_\theta}{\bar{q}} + \frac{V_\theta{}^2\cos\gamma}{r} - \frac{\bar{\omega}(rV_\theta)\cos\gamma}{r} + \frac{(\bar{\omega}r)^2\cos\gamma}{r} \tag{31}$$

Eqs. (17), (28), and (31) are combined into the final form of the momentum equation.

$$\frac{1}{2}\left(\frac{\partial V_m{}^2}{\partial \bar{q}}\right) = D(\bar{q}) + E(\bar{q})V_m{}^2 \tag{32}$$

where

$$D(\bar{q}) = \frac{\partial I}{\partial \bar{q}} - h\,\frac{\partial(\bar{S}/c_p)}{\partial \bar{q}} - (V_\theta - \bar{\omega}r)\left[\frac{1}{r}\,\frac{\partial rV_\theta}{\partial \bar{q}} + \frac{\bar{\omega}(rV_\theta)\cos\gamma}{r}\right] \tag{33}$$

$$E(\bar{q}) = \frac{\cos(\gamma+\phi)}{r_m} + \frac{\sin(\gamma+\phi)}{1 - Mm^2}\left\{\frac{\tan(\gamma+\phi)}{r_m} + \sec^2(\gamma+\phi)\,\frac{\partial \phi}{\partial \bar{q}} - \frac{1}{\lambda}\frac{\partial \lambda}{\partial m} - \left[1 - M_\theta{}^2 + M_\theta\left(\frac{\bar{\omega}r}{a}\right)^2\right]\frac{\sin\phi}{r}\right\} \tag{34}$$

For generality (and because it is useful for turbines), the equation was derived in a rotating coordinate system. If one applies it in a blade free space - which is the only totally legitimate situation - the equation is most readily used in a stationary coordinate system. It is put into that form simply by writing H for I and setting $\bar{\omega} = 0$. The $D(\bar{q})$ and $E(\bar{q})$ coefficients become

$$D(\bar{q}) = \frac{\partial H}{\partial \bar{q}} - h\,\frac{\partial(\bar{S}/c_p)}{\partial \bar{q}} - \frac{V_\theta}{r}\,\frac{\partial rV_\theta}{\partial \bar{q}} \tag{35}$$

$$E(\bar{q}) = \frac{\cos(\gamma+\phi)}{r_m} + \frac{\sin(\gamma+\phi)}{1 - Mm^2}\left[\frac{\tan(\gamma+\phi)}{r_m} + \sec^2(\gamma+\phi)\,\frac{\partial\phi}{\partial \bar{q}} - \frac{1}{\lambda}\frac{\partial \lambda}{\partial m} - (1 - M_\theta{}^2)\,\frac{\sin\phi}{r}\right] \tag{36}$$

No attempt has been made to transform the static enthalpy term which occurs in coefficients (33) and (35). One could write Eq. (25) as:

$$h = I - \frac{1}{2}V_m{}^2 - \frac{1}{2}V_\theta{}^2 + (\bar{\omega}r)V_\theta$$

and thereby recombine the terms in the coefficients. If the equations were truly to be solved as classical differential equations this would make sense; for a computer solution the form given is perfectly adequate.

THE MOMENTUM EQUATION - THE ANALYSIS PROBLEM

For the analysis problem it is assumed that the air angle distribution can be specified as a function of the $\bar{q}$- distance at each computing station. It is expedient, therefore, to write the momentum equation to suit the situation.

In Eq. (25), one may replace W_θ by $V_m\tan\beta$ where β is the relative air angle. The $\bar{q}$- derivative of such a formulation of Eq. (25), combined with Eq. (30), gives:

$$\frac{1}{\rho}\frac{\partial p}{\partial \bar{q}} = \frac{\partial I}{\partial \bar{q}} - h\,\frac{\partial(\bar{S}/c_p)}{\partial \bar{q}} - \frac{(1+\tan^2\beta)}{2}\,\frac{\partial V_m{}^2}{\partial \bar{q}} + V_m{}^2\tan\beta\,\frac{\partial \tan\beta}{\partial \bar{q}} + \frac{(\bar{\omega}r)^2\cos\gamma}{r} \tag{37}$$

This can be combined with Eq. (17), modified to recognize that

$$\frac{V_\theta{}^2\cos\gamma}{r} = \frac{V_m{}^2\tan^2\beta\cos\gamma}{r} + \frac{2(\bar{\omega}r)V_m\tan\beta\cos\gamma}{r} + \frac{(\bar{\omega}r)^2\cos\gamma}{r}$$

The result becomes:

$$\frac{1}{2}\,\frac{\partial V_m{}^2}{\partial \bar{q}} = F(\bar{q}) + G(\bar{q})V_m + J(\bar{q})V_m{}^2 \tag{38}$$

where

$$F(\bar{q}) = \cos^2\beta \left[\frac{\partial I}{\partial \bar{q}} - h\,\frac{\partial(\bar{S}/c_p)}{\partial \bar{q}}\right] \tag{39}$$

$$C(\bar{q}) = \cos^2\beta \left[\frac{2(\bar{\omega}r)\tan\beta\cos\gamma}{r}\right] \tag{40}$$

$$J(\bar{q}) = \cos^2\beta \left[\tan\beta\,\frac{\partial\tan\beta}{\partial\bar{q}} - \frac{\tan^2\beta\cos\gamma}{r} - \frac{\cos(\gamma+\phi)}{r_m} - \frac{\sin(\gamma+\phi)}{V_m^2}\left(V_m\,\frac{\partial V_m}{\partial m}\right)\right] \tag{41}$$

The $V_m(\partial V_m/\partial m)$ term, of course, comes directly from Eq. (28). The equivalent form for stationary coordinates can be obtained by setting ($I = H$, $\bar{\omega} = 0$). Additionally, for the stationary coordinates system, the relative angle, β, becomes the absolute air angle α.

SOLUTION TECHNIQUES AND CONVERGENCE

A few preliminaries are in order before any extended discussion of solution techniques for the combined momentum, energy and continuity equations. The first of them, which we will deal with only briefly, has to do with the specification of losses. As derived and written in the previous sections, the entropy, $\bar{S}$, (or, alternatively, $\bar{S}/c_p$) was left unmodified in the momentum equation. The entropy can be expressed in terms of pressure and temperature by

$$e^{-(\bar{S}/c_p)} = \frac{(P_o/P_i)^{\frac{\gamma-1}{\gamma}}}{T_o/T_i} \tag{42}$$

where, for the moment, P_o and T_o are pressures and temperatures at the current working point and P_i and T_i express conditions at the reference stations (upstream of a blade row, or of an entire turbomachine). The entropy is, of course, an entropy change from the value assigned at the reference station.

It is conventional practice - and it is so for the computer program which models our discussion - to permit losses to be specified in a number of alternative ways, i.e., isentropic efficiency, polytropic efficiency, loss coefficient, etc.

In all of these forms the same pressure and temperature terms as those in Eq. (42) occur. It is evident that efficiencies can be directly converted into values of entropy change, and that loss coefficients can be likewise treated as soon as an interim value of the velocity has been chosen at any stage of the iterative routine. In the design process, values of efficiency or of loss coefficient are directly specified as input by the designer; in the analysis process, it is possible (as we will detail below for the case of the axial compressor) to set up the computing process so that losses are obtained from a separate computing program subroutine which reflects an analytic or experimental formulation of blade section loss as a function of Mach number, incidence angle, etc.

In order to frame, and to make more intelligible, the main body of the discussion of the present section, we will briefly outline the steps necessary to obtain a converged solution for a given case. Two particular aspects of the solution procedure will, thereafter, be considered in greater detail.

General Description of Computing Routine

The overall discussion of the solution procedure will be given as though we are discussing a problem of analysis for a multistage axial compressor.

1. In the analysis problem, the computer is told as input the geometrical details of the compressor, i.e., the locii of the inner and outer walls of the machine, and the description of the blading of each blade row. It is told, also, the location and orientation of each computing station. It is instructed to obtain a solution for a specified flow and rotative speed, with a specified number of streamlines.

2. The first step in the computing process involves the initial placement of the streamlines. The usual first assumption is to place them so that the annulus area is divided in accordance with the number of streamtubes.

3. Having chosen the initial locus of each streamline, the computer is in a position to establish the radius of curvature of the streamline and the ϕ- angle at each computing mesh point, i.e., at each junction of a prescribed computing station and a streamline.

4. Each iterative pass is made through the entire flowpath, starting at the first computing station and proceeding to the last, with no change to the streamline pattern. For illustrative purposes, we assume that the routine has proceeded to, say, the third stator discharge. We will verbally carry it on to the discharge of the fourth rotor.

5. Generally, it is expedient to place computing stations at the leading and trailing edges of each blade row, though this becomes unnecessary and costly of computing time for closely spaced blade rows in the rear of a multistage compressor. Since we are assuming that computations for the current iterative pass have been completed at the stator discharge, all velocities, pressures, etc., are known at that station for each streamline. The computer

simply presumes constancy of angular momentum in order to move across the blade gap to get to the next station (the leading edge of the fourth rotor, in our illustration). Under these circumstances, it will use Eq. (32) written as for a stationary coordinate system as the appropriate momentum equation, since it can readily determine all terms except, as yet, the Mach number terms and the static enthalpy.

6. An initial estimate is made of velocities at each computing mesh point at the station. Generally, the estimated values are those obtained at the same station for the (n-1)th iterative pass. The estimated velocities immediately make it possible to establish values of the velocity dependent terms, i.e., the static enthalpy and the Mach numbers.

7. Using the mid streamline velocity value as a starting point (i.e., as an integration constant) the momentum equation is solved in a step-wise fashion, giving new values of velocity for the inner and outer halves of the annulus. The new velocities make it possible to establish values of density.

8. The mass flow across the annulus is computed with the continuity equation. In general, the mass flow will not satisfy the mass flow desired. The mid streamline velocity is increased or decreased as required, and the process is repeated until continuity is satisfied. As a final step, continuity is applied for each streamline. In general, the streamtubes will not contain their proper percentage of flow. The computer determines, and stores, an error function for streamline location. (It is to be noted the real process is somewhat more complicated than is implied by the above cursory description. Details of the full process, and of some of its niceties, are given below.)

9. The computer now has available to it all velocities, angles, Mach numbers, etc. at the leading edge of the fourth rotor (continuing with our example). It, therefore, can determine conditions relative to the rotor, i.e., incidence angles, Mach numbers, etc. It also has values from the (n-1)th iterative pass of the conditions downstream of the rotor. It has, in short, all that is needed to obtain the rotor downstream air angle, and the rotor downstream entropy. This it does from a cascade subroutine, the details of which are discussed later.

10. The momentum equation is applied, and the same process as before is used to establish conditions at the rotor discharge.

11. In this fashion, the computing process continues to the last computing station. The previously determined streamline location error functions are now simultaneously applied to redefine the entire streamline pattern throughout the machine, and the (n+1)th iterative pass is begun. (Once again, the details of the streamline shift are to be elaborated later in this section.)

Simultaneous Solution of the Momentum and the Continuity Equation

We will devote some attention to the problems attendant to the simultaneous solution of the momentum and the continuity equation at any one station, for any one iterative pass, for which the streamline locations are treated as fixed.

It is considered worthwhile to document this aspect of the solution routine. It has received very little attention in the open literature and it can make a significant difference in the computing time required for a given problem. We rely, here, principally on Ref. 9 which is an unpublished communication from R. M. Hearsey, and on Ref. 10 which is the overall description of the computing program we are specifically discussing.

Streamline curvature computing techniques are characterized by the fact that the momentum equation and the continuity equation are applied successively. One chooses a velocity at some point in the passage (usually at mid-passage). A step integration of the momentum equation defines velocities across the passage consistent with the original choice at midpassage. This interim knowledge of velocities (along with the entropy) then permits the computation of static densities, and, therefore, of the application of the continuity equation. In general, of course, the initial choice of a midpassage velocity will be incorrect.

The question to which Hearsey addresses himself in Ref. 9 and Ref. 10 is: After any one station iteration, when the degree to which continuity has not been properly satisfied is established, is there a logical way to maximize the possibility that the next choice of midpassage velocity will be the correct one? The question is particularly an important one when the passage Mach numbers are near unity. The answer is to derive an explicit, albeit an approximate, expression for the rate of change, with respect to the midpassage velocity level, of mass flow across the annulus at any one computing station.

The total mass flow across the passage is:

$$\bar{W} = \int \rho V_m \, dA$$

where the integration is from the hub to the case, and A is the area projection normal to the meridional velocity, V_m. For the design case, where the total temperature, T, and the tangential component of velocity, V_θ, are fixed, we write:

$$\bar{W} = \int V_m \rho_T \left(1 - \frac{V_m^2 + V_\theta^2}{2gJc_p T} \right)^{\frac{1}{\gamma - 1}} dA$$

where ρ_T is the total density and the bracketed-term is the perfect gas ratio of static-to-total density.

A differentiation with respect to $(V_m)_{mid}$, i.e., with respect to the midpassage meridional velocitiy, yields

$$\frac{d\bar{W}}{d(V_m)_{mid}} = \int \rho \left[\frac{dV_m}{d(V_m)_{mid}}\right] \left(1 - Mm^2\right) dA$$

where $Mm = V_m/a$ and a is the local sonic velocity.

Hearsey makes the admittedly incorrect, but heuristically justified, assumption that

$$dV_m/d(V_m)_{mid} \approx V_m/(V_m)_{mid}$$

This is equivalent to assuming that the ratio of V_m to $(V_m)_{mid}$ remains constant when $(V_m)_{mid}$ is varied. This cannot be so, but it is a reasonable approximation for small changes. The result is then

$$\frac{d\bar{W}}{d(V_m)_{mid}} = \frac{1}{(V_m)_{mid}} \int \rho V_m (1 - Mm^2) dA \tag{43}$$

For the analysis case the relative flow angle is held fixed. The corresponding expression for density is:

$$\rho = (\rho_T)_R \left[1 - \frac{V_m^2(1 + \tan^2\beta)}{2gJc_pT_R}\right]^{\frac{1}{\gamma - 1}}$$

where T_R is the relative total temperature, $(\rho_T)_R$ is the relative total density, and $V_m^2(1 + \tan^2\beta)$ is W^2, the relative velocity. In a fashion very similar to that described above, one obtains

$$\frac{d\bar{W}}{d(V_m)_{mid}} = \frac{1}{(V_m)_{mid}} \int \rho V_m (1 - M_R^2) dA \tag{44}$$

where $M_R = W/a$.

We have been leading up to a point which is certainly not new, but which has received surprisingly little discussion in the literature. No one questions the relation between mass flow rate and velocity at, for example, the throat of a one-dimensional nozzle. The flow rate has a maximum when the velocity is sonic; the flow rate decreases as velocity decreases in the subsonic regime, and decreases as velocity increases in the supersonic regime.

In a non-one-dimensional nozzle (i.e., in any flow passage where two-, or three-, dimensional effects exist) it must be recognized that the "sonic" point (i.e., the point at which maximum flow is reached) is one at which some streamtubes are supersonic while others are subsonic. At, for example, the throat of a real turbine nozzle, the radial distribution of blade twist and entropy level may, via the momentum equations, impose a significant radial velocity gradient. Maximum flow, or "choke", at the nozzle throat can well correspond to supersonic flow through some streamtubes, and subsonic through others.

The present Lecture is dealing with axisymmetric flows, and not directly with situations like that near a turbine nozzle throat, (where the picture is further complicated by cross-channel velocity gradients). It remains true, however, that almost never in a real situation will a "choke", or a maximum flow, situation be truly one-dimensional. A recognizeable axisymmetric example of the situation we are discussing is that of a plug nozzle with significant sidewall curvature in the region of the throat. The point of maximum flow through such a nozzle occurs with some streamtubes at a subsonic level and others at a supersonic level. Expressions (43) and (44) represent attempts to deal with the problem in the context of a computing system such as we have been discussing.

The situation we have discussed makes the computing process to be a tricky one, most especially when the flow specified is close to the maximum possible flow which can be passed at the computing station in question, and the computing process is at an interim point where the streamline pattern is still incorrect. To indicate how it may be dealt with, Steps 6, 7, and 8 of the previously given overall description of the computing process will be restated in greater detail. We consider a solution on the subsonic side of maximum flow, but close to maximum flow. The description would be completely symmetric if a supersonic solution were asked for.

1. At the beginning of the iterative loop at any one computing station, first estimate the meridional velocity profile. Typically, one starts by assuming some uniform value, or the values resulting from the (n-1)th major iterative pass if there was one.

2. Determine a solution of the momentum equation consistent with $(V_m)_{mid}$, the meridional velocity value assumed at the mid radius point. A number of iterative loops is implied here, since the values of meridional velocity initially guessed at at radii other than mid radius (to get the process started) are not, in general, consistent.

3. Determine the flowrate, and the rate of change of flowrate with $(V_m)_{mid}$ (using either Eq. (43) or (44)).

4. If $d\bar{W}/d(V_m)_{mid}$ is negative, this means the initially chosen $(V_m)_{mid}$ is on the supersonic side of the $(V_m)_{mid}$ for maximum flow. In this event, since we wish a subsonic solution for the case being discussed, it is necessary to move as rapidly as possible toward the subsonic region. This is accomplished by letting $\Delta V_m = -0.1(V_m)_{mid}$, i.e., by setting the next iteration choice of $(V_m)_{mid}$ to be 0.90 of the previous choice.

5. If $d\bar{W}/d(V_m)_{mid}$ is positive, the ΔV_m to be applied to the next iteration is given by

$$\Delta V_m = (\bar{W}_{specified} - \bar{W}_{computed}) \;/\; [d\bar{W}/d(V_m)_{mid}]$$

6. At all radii, take

$$(V_m)_{new} = (V_m)_{old} + \Delta V_m$$

for the starting choice of the next iteration, and repeat, starting at Step 2. Usually, a maximum of about twenty such minor iterations are required to satisfy continuity to within some tolerance, typically 1 part in 1,000 to 1 part in 10,000, and the meridional velocities at all radii repeat to within the same tolerance. The tolerance level is a user input choice.

7. Additional restrictions are placed upon ΔV_m. It is restrained so that after incrementing by ΔV_m, the midradius V_m will be neither higher than it was for any profile previously determined to be supersonic in nature, nor lower than for any profile that was a subsonic solution, and yielded less than the specified flow. Thus, the new $(V_m)_{mid}$ will be restrained to a range of plausible values. Further, if the new midradius V_m is restrained from being within a small amount of the limits, rather than being allowed to equal the limiting values, the plausible range will be continually refined, i.e., narrowed. This is particularly useful in cases where, because the overall streamline pattern is not yet accurate, the specified flow rate is greater than the maximum possible. In this case the ΔV_m's determined will result in repeated inadvertant transfers from one branch of the continuity equation to the other, followed by intentional switches back to the defined branch. After a number of iterations through the procedure the final midradius V_m will be a value very close to that corresponding to the maximum flow rate, which is the closest possible to a solution at this point in the development of the overall solution. (Since the overall streamline pattern, in general, is not yet properly converged to its true location.)

Two other points should be noted in regard to the continuity solution:

1. The procedure described works just as well for situations which are completely subsonic, partly subsonic and partly supersonic on the subsonic branch, partly subsonic and partly supersonic on the supersonic branch, or fully supersonic.

2. Solutions requested which are too close to the maximum flow can also get into trouble because of the $(1 - M_m^2)$-term which occurs in the denominator of Eq. (28). In the axisymmetric computer program we have been describing, the value of this term is restricted to the range 1.00 - 0.02. This, obviously can introduce inaccuracies for computations at very high Mach number.

Marsh, in Ref. 11, clearly presents and comments on the problem we have been discussing. He concludes, however, that only solutions which are wholly subsonic or wholly supersonic are possible with the streamline curvature approach. Hearsey and Marsh discuss the matter further in Ref. 12; Hearsey's argument is substantially that presented above.

Convergence Techniques

One of the reasons why a streamline curvature computing procedure can be practically attractive is that, properly executed, it is capable of surprisingly rapid convergence, even for annular, bladed, passages of complex shapes. There are, however, differences in computer routines in this regard. It is known, for example, that two essentially similar programs exist which have been tested against each other on an analysis problem involving a multistage axial compressor. One of the programs took more than twice as long as the other to obtain a converged solution, even though the two converged solutions were quite close.

Refs. 10, 13, and 14, deal with the problem. Since Ref. 14, by Wilkinson, is publicly available, we will base the discussion chiefly upon it.

In the previous sections an overall description was given of the successive application of the momentum equations and the continuity equation in the effort to obtain a converged solution. The result of any one iteration produces a set of new streamline radii. If these new radii are introduced directly into the solution for the next iteration, it is often - most often - the case that the process will diverge rather than converge. A relaxation factor is needed to control the process. It is formulated in the context of the following formula.

$$r_n = r_o + R(r_p - r_o) \tag{45}$$

where

r_n = the damped new value of streamline radius to be used in a succeeding iteration.

r_o = the radius which was used on the previous, or "old" iteration.

r_p = the "predicted" radius which resulted as an undamped value from the previous iteration.

R = the relaxation factor.

It is principally the purpose of the remainder of the present section to indicate a logical basis for establishing the relaxation factor. The method we follow is that, simultaneously, of Hearsey and Wilkinson.

We examine a uniform axial flow in a constant area, annular, duct (Fig. 3). It is assumed that no entropy or enthalpy gradients exist. Radial computing stations are uniformly spaced Δx apart. We consider, first, a case involving a fixed absolute whirl velocity. The momentum euqation, Eq. (32), becomes (with $\bar{q} = r$):

$$\frac{1}{2}\frac{\partial V_m^2}{\partial r} = \frac{V_m^2}{r_m} - \frac{V_\theta}{r}\frac{\partial}{\partial r}(rV_\theta) \qquad (46)$$

where the terminology is as previously defined.

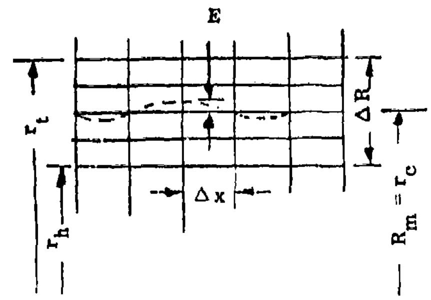

Figure 3

We assume but small deviations from mid-radius conditions and approximate Eq. (46) by:

$$\frac{dV_m}{dr} = \frac{(V_m)_m}{r_m} - T \qquad (47)$$

where $(\ )_m$ indicates mid-radius conditions and R_m is the mid radius. Also:

$$T = \left[\frac{V_\theta}{(V_m)_m}\right]\frac{1}{r}\frac{d}{dr}(rV_\theta)$$

An error E in streamline location is assumed at one station only. The radius of curvature at that station becomes:

$$r_m = -\frac{\Delta x^2}{E}$$

where the negative sign arises from the chosen convention of a calling a positive r_m to apply to a convex-upward curve.

Eq. (47) becomes:

$$\frac{dV_m}{dr} = \frac{2E(V_m)_m}{\Delta x^2} - T$$

or, integrating:

$$V_m(r) = (V_m)_m - \left\{\frac{2E(V_m)_m}{\Delta x^2}\right\} - T\ (r - R_m) \qquad (48)$$

The mass flux distribution across the passage is approximated by:

$$\rho V_m = (\rho V_m)_m + (\rho)_m\left[V_m - (V_m)_m K\right] \qquad (49)$$

where

$$K = \frac{1}{(\rho)_m}\left[\frac{d(\rho V_m)}{dV_m}\right]_m$$

The specific flow between any two radii, r_1 and r_2, can be expressed (using Eqs. (48) and (49)) as:

$$\int_{r_1}^{r_2} V_m dr = (\rho V_m)_m \left\{(r_2 - r_1) + \frac{1}{2}\left(\frac{2EK}{\Delta x^2} + T\right)\left[(r_2 - R_m)^2 - (r_1 - R_m)^2\right]\right\} \qquad (50)$$

In order to determine the effect of the error, E, a comparison is made between the streamline predicted location when the error is E and the location when E = 0, i.e., its correct location. The predicted new location, r_p, can be found by solving the following equation for r_p:

$$\int_{r_h}^{r_p} \rho V_m dr = 0.5 \int_{r_h}^{r_t} \rho V_m dr$$

where subscripts h and t refer, respectively, to the hub and tip.

Using Eq. (50), and assuming that $r_p - r_h \approx R_m - r_h$, one obtains:

$$r_p = R_m - \left(\frac{KE}{\Delta x^2} + \frac{T}{2(V_m)_m}\right) \frac{\Delta R^2}{4}$$

where $\Delta R = r_t - r_h$.

For E = 0, the correct location is given by:

$$r_c = R_m - \frac{T \Delta R^2}{8(V_m)_m}$$

and the old location (in the sense of Eq. (45) is:

$$r_o = r_c + E$$

The shift in streamline position, from one iteration to the next as a result of the error E is:

$$r_p - (r_c + E) = -E\left(1 + \frac{K \Delta R^2}{4 \Delta x^2}\right)$$

Application of Eq. (45) gives, therefore, (with $r_n = r_c$):

$$r_n = r_c + E - R\left[-\frac{KE}{4}\left(\frac{\Delta R}{\Delta x}\right)^2 - E\right]$$

or

$$R = \frac{1}{1 + \frac{K}{4}\left(\frac{\Delta R}{\Delta x}\right)^2}$$

The case considered with an imposed error at one point only would produce the largest curvatures and the maximum shift in radius from one pass to the next. The relaxation factor, therefore, is a minimum. The other extreme can be considered to be the case where the same error is imposed on all streamlines, producing no curvature. In this event, no damping would produce the most rapid convergence. Both Wilkinson and Hearsey recommend an "optimum" damping which is the mean of the "short wave" case and that of the very "long wave" case, where R = 1. Thus,

$$R = \frac{1}{1 + \frac{K}{8}\left(\frac{\Delta R}{\Delta x}\right)^2} \tag{51}$$

We must still evaluate K.

$$K = \frac{1}{(\rho)_m}\left(\frac{d\rho V_m}{dV_m}\right)_m$$

For a perfect gas

$$\rho V_m = \rho_T V_m \left[1 - \frac{V_m^2 + V_\theta^2}{2gJc_p T_T}\right]^{\frac{1}{\gamma - 1}} \tag{52}$$

where

ρ_T = total density,

T_T = total temperature

The only variable (for our case) in Eq. (52) is V_m. We differentiate with respect to V_m to get:

$$\frac{1}{(\rho)_m}\left(\frac{d\rho V_m}{dV_m}\right)_m = (1 - M_m^2)_m$$

where M_m is the Mach number based on the meridional velocity. Eq. (51) then becomes:

$$R = \frac{1}{1 + \frac{(1 - M_m^2)}{8}\left(\frac{\Delta R}{\Delta x}\right)^2} \quad (53)$$

The analysis when relative flow angle, rather than V_θ, is fixed is very similar. Here, however, it is to be recognized that total enthalpy varies as the meridional velocity varies. The applicable form of the momentum equation is:

$$\frac{d\,\frac{1}{2}V_m^2}{dr} = \frac{V_m^2}{r_m} + \frac{V_\theta}{r}\frac{d}{dr}(rV_\theta) + \frac{dH}{dr}$$

where H is the total enthalpy and

$$V_\theta = \omega r + V_m \tan\beta$$

$$H = H_i + (\bar{\omega} r V_\theta)_i + \bar{\omega} r(\bar{\omega} r + V_m \tan\beta)$$

where the i subscript refers to blade inlet conditions and β is the relative flow angle.

If we treat these relations precisely as above, the relaxation becomes

$$R = \frac{1}{1 + \frac{(1 - M_R)^2}{8}\left(\frac{\Delta R}{\Delta x}\right)^2 \cos^2\beta} \quad (54)$$

where M_R is the relative Mach number, the K-coefficient has been evaluated by the relation

$$\rho V_m = \rho_{TR} V_m \left[1 - \frac{V_m^2(1 + \tan^2\beta)}{2gJc_p T_R}\right]^{\frac{1}{\gamma - 1}} \quad (55)$$

where T_R is the relative total temperature and ρ_{TR} is the relative total density.

In practice, the constant of 8.0 is usually treated as an input variable which can be raised or lowered as the situation appears to demand. The relaxation factor does, however, appear realistically to account for the aspect ratio of the computing stations (i.e., ΔR/Δx) and for the flow Mach number level. The value of 8.0 for the constant gives excellent results for axial compressors; for turbines of high swirl levels, greater damping is often required.

Computation of Streamline Curvature

The axisymmetric computer program which is being described gives the user an option of using a spline method to establish the curvature values, or a simple finite difference method (three points). By experiment, it has been determined that the answers obtained with the two methods are nearly identical, and that the finite difference approach reaches convergence considerably more rapidly and reliably.

Wilkinson more thoroughly investigated the problem in Ref. 14 and arrives at the same conclusion. He says: "A large number of methods for finding the second derivative of a curve, and hence streamline curvature, have been analyzed to find their accuracy and stability characteristics. It has been shown that, in this context, finite difference methods are best and the conventional spline method the worst." No attempt will be made to reproduce Wilkinson's argument; Ref. 14, is, however, strongly recommended as required reading for anyone heavily involved in the streamline curvature approach.

The curvature computation (at any Station I) comes from the standard formula.

$$\frac{1}{r_m} = \frac{\frac{d^2r}{dz^2}}{\left[1 + \left(\frac{dr}{dz}\right)^2\right]^{1.5}} \quad (56)$$

where

$$\frac{dr}{dz} = \frac{1}{2}\left(\frac{r_{I+1} - r_I}{z_{I+1} - z_I} + \frac{r_I - r_{I-1}}{z_I - z_{I-1}}\right) \quad (57)$$

$$\frac{d^2r}{dz^2} = \left(\frac{2}{z_{I+1} - z_{I-1}}\right)\left(\frac{r_{I+1} - r_I}{z_{I+1} - z_I} - \frac{r_I - r_{I-1}}{z_I - z_{I-1}}\right) \quad (58)$$

AN EMPIRICAL CASCADE LOSS ROUTINE FOR AXIAL COMPRESSORS

We have, so far, discussed the momentum equations and solution techniques with some generality. They could as well be used for turbines as for compressors. From this point on, in this Lecture, the discussion will be directed specifically to the axial compressor.

Unlike the situation which exists with regard to the turbine, there is a considerable body of public data which permit realistic assessment of the loss and turning which will result from a compressor cascade of prescribed geometry, and with prescribed upstream and downstream velocities, or Mach numbers.

The conventional approach, at least for lower Mach number profiles, is to base a cascade subroutine upon the considerable data presented in NASA SP-36 (Ref. 15). Since this report has become a virtual reference text in the axial compressor industry, it will be unnecessary for us to do more than to indicate how its data can be computerized, and to indicate the comparatively few areas where its approach can be supplemented, or modified.

It is most common throughout the industry that compressor blading is specified (in terms of camber, stagger, thickness and section type) on cylindrical sections at a number of radii along the blade. Except for the latter day very high Mach number fans, the sections are generally NACA-65 Sections, some NGTE Parabolic-Arc Sections or the Double Circular Arc. It is of interest to note that the frequent use of these sections rests more with the availability of a broad range of loss and turning angle data than it does from proven superiority to other possible sections.

Since the streamline does not, in general, lie on a cylinder, it is first necessary to transform the given cascade geometry on cylinders to that along stream surfaces. The standard method used is that of Ref. 16.

The angle the blade edges make with the radial directions when viewed from upstream parallel to axis is designated as the angle ε_1, the lean angle. The streamline slope ($\tan\phi$) at any point is given from the streamline radii established at any point in the computing process. The effective metal angles as seen from the stream surface direction are given by:

$$\tan\bar{\beta}_s = \frac{\cos(\gamma + \phi)}{\cos\gamma}\tan\bar{\beta}_c - \sin\phi\tan\varepsilon_1 \tag{59}$$

where

$\bar{\beta}_s$ = the blade angle in the streamline direction

$\bar{\beta}_c$ = the blade angle on a cylindrical surface

The angles ϕ, γ and ε_1 are as previously defined.

The blade section effective camber angles are determined as the difference between the $\bar{\beta}_s$-values at leading and trailing edges. Assuming the mean of the inlet and outlet radii to apply, the solidity, thickness/chord ratio and the maximum camber point for each section are then interpolated from the given input data.

The solidity and thickness/chord ratio are corrected for the streamline slope using

$$\sigma_s = \sigma_c/\cos(\phi)m$$

$$(t/c)_\sigma = (t/c)_c\cos(\phi)m$$

where

σ = the solidity
t/c = the thickness/chord ratio
$(\phi)m$ = the mean of inlet and outlet streamline slope angles

Thus, the geometry of each cascade section along a streamline has been specified in terms of blade outlet angle, camber, point of maximum camber, solidity and thickness/chord ratio.

It is well to indicate the overall method. In the context of the convergence routine described earlier, a means is required to establish the loss across a cascade, and the turning produced by the cascade, every time the convergence routine establishes an interim (not yet converged) set of values of air inlet angle and Mach number at a station at the leading edge of a blade. At this point, the computing process must, as it were, be directed to the cascade subroutine in order that it be given the information needed to determine the angular momentum change across the cascade and the entropy change across the cascade. The cascade subroutine provides, directly, the means by which values of deviation angles and loss coefficient are determined.

In regard to the deviation angle, one obvious approach is to assume the total deviation (δ) to be a summation of a low speed deviation (δ_P), additional deviation arising when the inlet Mach number exceeds the critical Mach number, (δ_M), additional (positive or negative) deviation due to cascade axial velocity ratios other than unity (δ_{VA}), additional deviation due to operation at other than optimum incidence (δ_I) and, finally, any additional deviation imposed as input (δ_A). Once again, a reliance on Ref. 15 strongly pushes one toward an approach of this sort.

In a directly analogous fashion, loss coefficient can be considered as a summation of low speed minimum loss coefficient (ω_{min}) plus an increment due to supercritical operation (ω_M), plus some increment attributable of off-design operation (ω_{OD}) plus an arbitrarily imposed additional loss (ω_T).

For both deviation and loss at values of incidence other than that at minimum loss, some reference positive and negative stall incidence values are needed. It has been current practice since the time when the earlier English investigators began to study the problem, to define positive and negative stall incidence values as those at which the loss level reaches twice the value at minimum loss. This is arbitrary, but not as much so as it sounds. Experimentally, actual stall and separation occur shortly after the double-loss point for most cascades.

Deviation Angle

The reference minimum-loss deviation angle can be determined from:

$$\delta_R = (\delta_o)_{10}(K_\delta)_t(K_\delta)_{SH} + \frac{m\theta}{\sigma^b} \tag{60}$$

which is a combination of Eqs. 269 and 271 of Ref. 15. The deviation can be made additionally a function of the point of maximum camber, as suggested by Carter and Hughes in Ref. 17.

In Eq. (60):

δ_R is the reference minimum loss deviation angle

$(\delta_o)_{10}$ is the reference zero-camber minimum loss deviation angle as given by Fig. 161 of Ref. 15, as a function of the upstream air angle and the solidity

$(K_\delta)_t$ is the maximum-thickness correction factor as given by Fig. 172 of Ref. 15

$(K_\delta)_{SH}$ is the blade section shape correction factor for minimum loss deviation angle (e.g., Ref. 15 suggests 1.0 for NACA-65 and 0.7 for DCA-sections)

σ is the cascade solidity

b is the solidity exponent as given by Fig. 164 of Ref. 15 as a function of upstream air angle

m is the slope factor for the deviation expression given for NACA-65 sections as a function of air angle and solidity on Fig. 162 of Ref. 15, and for DCA-sections on Fig. 168.

The additional deviation attributable to the meridional velocity ratio, δ_{VA}, can most easily be determined by a simple formulation proposed by Horlock in Ref. 18.

$$\delta_{VA} = 10\left(1 - V_{m_2}/V_{m_1}\right) \tag{61}$$

where

V_{m_2}/V_{m_1} = the meridional velocity ratio across the cascade section. It is to be noted that the cascade outlet velocity is not known for any (n)th iteration. Values from the (n-1)th iteration are used to evaluate δ_{VA}, and are continuously updated.

In order to arrive at a figure for the additional deviation attributable to operation of the cascade at supercritical conditions, it is first necessary to establish the critical Mach number, i.e., the upstream Mach number at which local sonic velocity is reached on the cascade suction surface. Ref. 19 by Jansen and Moffatt provides a ready means for this determination. It was noted that the pressure coefficient corresponding to the minimum pressure point on the blade suction surface remains essentially unchanged up to the point at which critical Mach number is reached. Ref. 19 suggests, therefore, that a knowledge of the incompressible pressure distribution on the blade will immediately provide means for establishing the critical Mach number. Thus,

$$\left(\frac{\Delta P}{q_1}\right)_{incomp} = \left(\frac{\Delta P}{q_1}\right)_{comp} = \left(\frac{V_{max}}{V_1}\right)^2 - 1$$

$$= \frac{1 - \left[\frac{2}{\gamma + 1} + \frac{\gamma - 1}{\gamma + 1} M_{1C}^2\right]^{\frac{\gamma}{\gamma - 1}}}{\left(1 + \frac{\gamma - 1}{2} M_{1C}^2\right)^{\frac{\gamma}{\gamma - 1}} - 1} \tag{62}$$

Ref. 19 suggests a simple expression for the incompressible ratio of maximum surface velocity to inlet velocity, which is deduced from an inspection of NACA low-speed cascade data. It is a function of solidity and cascade turning. Thus,

$$\frac{V_{max}}{V_1} = 1 + E\left(\frac{\Delta V_\theta}{\sigma V_1}\right) + F \tag{63}$$

where

$E = 0.4 + t/c$
$F = 0.03 + 0.7(t/c)$

For a value of V_{max}/V_1 for a given cascade, Eq. (62) can be solved iteratively for the critical Mach number. Fig. 4, taken from Ref. 19, shows a comparison between the critical Mach number computed in this fashion and experimental data on cascade turning as a function of Mach number taken from Ref. 20.

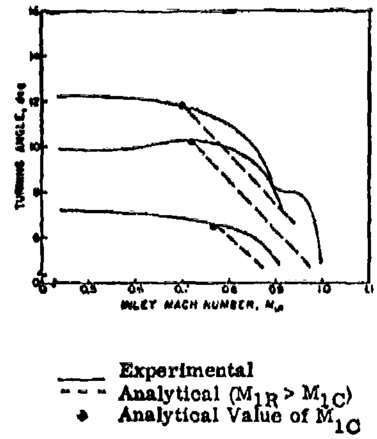

Figure 4

In the Appendix of Ref. 19, it is suggested that a correction (δ_M) be applied to design turning when the Mach number exceeds critical. It is based on the supposition that supercritical operation is accompanied by a shock on the suction surface which causes separation. When no reattachment is presumed the separation interface is a free streamline along which the velocity is constant. Thus the velocity ratio across the cascade does not exceed the value corresponding to that at critical Mach number. Ref. 19 establishes a value of δ_m based on this assumption. The dotted lines on Fig. 4 show the results for these cascades. The correspondence is creditable. It does not, however, appear to apply at all to a rotating cascade (where critical Mach number is most often exceeded), probably because of boundary layer centrifugation and, hence, reattachment.

Fig. 5 has been reproduced from Ref. 21 by Horlock. It shows an example of what is referred to as a "Mellor" curve, since this form of plotting cascade information was first proposed by George Mellor. A number of successful approaches to establishing the additional deviation attributable to off-design operation (δ_T) are based, in one fashion or other, on the Mellor curves for NACA-65 profiles. Thus, for example, Fig. 177 of Ref. 15 is Lieblein's formulation (for the same data) of the rate of change of deviation with incidence at minimum loss incidence. It is the slope of the curves on Fig. 5 at minimum loss incidence. The slope at choke is evidently zero. The slope at positive stall can readily be synthesized in a fashion similar to Lieblein's formulation at minimum loss. With this information, a polynomial representation of the data on the Mellor curves can be synthesized.

It is necessary, of course, to be able to establish the minimum loss incidence angle, and also the incidence range to the arbitrarily defined (i.e., double loss) choke and stall incidence angles. Ref. 15 (Eqs. 261 and 262) gives the low speed minimum loss incidence angle as:

$$i_R = i_{0_{10}} (K_i)_t (K_i)_{sh} + n\,\theta \quad (64)$$

where

i_R is the reference (i.e., low-speed) minimum-loss incidence angle

$i_{0_{10}}$ is the zero-camber reference minimum-loss incidence angle, (Ref. 15, Fig. 137)

$(K_i)_t$ is the blade thickness correction factor for zero-camber reference minimum-loss incidence angle, (see Ref. 15)

$(K_i)_{sh}$ is the blade section shape correction factor for zero-camber reference minimum-loss incidence angle, (see Ref. 15)

n is the slope factor (Ref. 15, Fig. 138)

and

θ is the blade section camber angle.

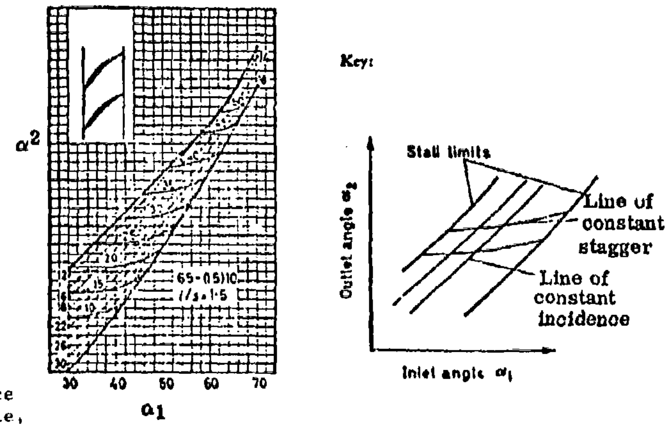

Figure 5

Fig. 6 has been reproduced from Ref. 22. It shows the incidence operating range from minimum loss to choke and from minimum loss to stall as a function of cascade geometry for, once again, the same NACA-65 cascade data. To make it applicable to a computer, the data represented by Fig. 6 must be analytically formulated; the precise fashion in which this is done is not important.

Loss Coefficient

The low speed minimum pressure loss coefficient once again, can be determined exactly as suggested in Ref. 15. The diffusion parameter at minimum loss incidence is determined from:

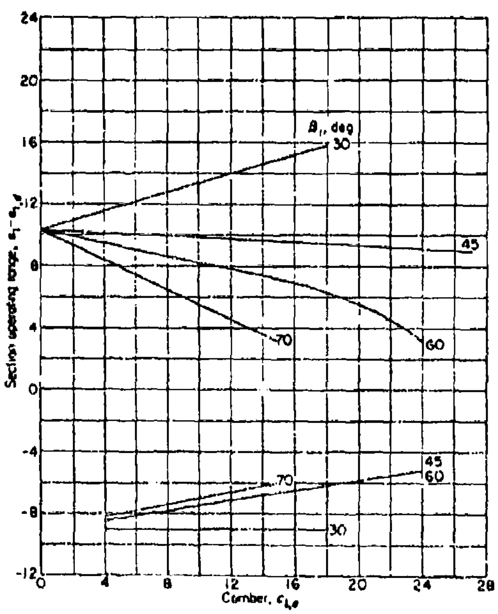

Figure 6

$$D = 1 - \frac{V_2}{V_1} + \frac{\Delta V_\theta}{2\sigma V_1} \tag{65}$$

where the velocities in Eq. (61) are relative to the cascade and σ is the cascade solidity.

Fig. 203b of Ref. 15 yields (as a function of D) a total pressure loss coefficient parameter, P; the low-speed minimum total pressure loss is obtained from:

$$\omega_{min} = 2P\ \sigma/\cos\beta_2 \tag{66}$$

where, at this point, β_2 is the low speed air leaving angle (at minimum loss incidence) relative to the cascade.

When the Mach number exceeds the critical Mach number (which is determined as described above), an increment of loss is added. The computation for the additional loss can be established essentially as described by Miller, Lewis and Hartmann in Ref. 24.

When the Mach number is above unity it is assumed that a normal shock extends across the passage. The impingement point of the shock on the suction surface is determined by the geometry of the cascade. It is assumed that the Mach number at the shock is taken to be that resulting from a Prandtl-Meyer expansion through the angle subtended by the suction surface upstream of the point of shock impingement. The shock loss is then calculated as it would exist for a Mach number which is the average of the inlet Mach number and that existing at the end of the Prandtl-Meyer expansion.

When the incident Mach number is above critical but less than unity, the same kind of computation as that described above is performed for an incident Mach number of unity. The loss coefficient, ω_M, can then be obtained by assuming a linear variation of loss coefficient with Mach number from a value of zero at the critical Mach number to the calculated value at a relative Mach number of unity.

The loss increment, ω_T, is based, once again, on NACA single-stage rotor test data which indicate a markedly higher loss coefficient for rotor tip sections, relative to levels which would be expected of the same sections at different radii. Fig. 203A, of Ref. 15 shows the effect. To allow for the inclusion of the empirically observed increment, the loss coefficient ω_T can be added. It is specified to be some multiple of ω_{min} at the tip and to be zero at the midspan point, with a cubic variation between. For the examples given later, the additional tip loss has been taken to be equal to the minimum loss, but only for the early stages of the multistage examples.

Fig. 7 shows, schematically, the change in the loss coefficient vs incidence angle as a function of the operating Mach number. The minimum loss at higher operating Mach numbers is increased, and the minimum loss incidence is increased. The range from choke to stall is decreased.

An inspection of the cascade data and single stage data contained in Ref. 15 and in Refs. 25 to 36 shows that a comparatively simple empirical variation of minimum loss incidence with Mach number can be deduced. It is of the form:

$$\frac{i_M - i_R}{\theta} = f(M) \tag{67}$$

where

i_M is the minimum loss incidence at actual Mach number

i_R is the low speed minimum loss incidence

θ is the section camber

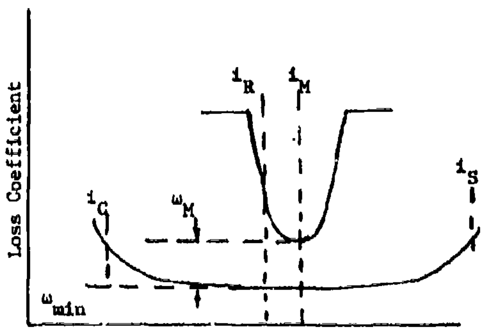

Figure 7

Finally, in order to determine the loss incurred by operating at incidence other than design, it is necessary to know the working range from minimum loss to stall, and that from minimum loss to choke. The percentage range decrease with Mach number can also be deduced from Refs. 15 and 25 thru 36. The magnitude of the range decrease is, for a low-speed section like the NACA-65:

M_1	.8	1.0	1.25
% WR	.712	.6	.5

The range decrease is the same for positive and negative incidence.

For DCA sections, the range percentage decrease is very nearly the same as that for a low speed section for negative incidence; for positive incidence, however, it is:

M_1	.8	1.0	1.25
% WR	.812	.8	.75

A good approximation to the actual test data could be obtained by using a polynomial fit of the numbers tabulated above.

One further point is to be noted. The loss horeshoes expressed as above are usually given by a polynomial curve. At incidence angles well away from design, the computed loss coefficient would become excessive. It is necessary, therefore, to limit it to some maximum value, both on the positive and negative sides of the minimum loss incidence. In the examples to follow, for example, the maximum was set to be 0.35.

BLOCKAGE

The ability to compute the flow through a multistage compressor depends heavily upon a means to establish the varying blockage through the annulus. Most of the published material dealing with end wall boundary layers in multistage axial compressors (e.g., Mellor, Horlock, L.H. Smith, Jr., Hirsch, etc.) is engaged in an important attempt to establish a model for the extremely complicated end wall flow. The first attempt to deal with the question on an overall basis was made by Jansen in 1967 (Ref. 37). He was quickly followed by Stratford. These attempts had a considerably more heuristic objective than much of the more recently published material. Jansen's concern, for example, was to establish a simple model with which to compute the overall blockage build-up in a multistage compressor. The concern was not with the details of the endwall flow, but with the overall computation of the performance of multistage compressors of arbitrary geometry.

Jansen simply postulated that the endwall boundary layer could be treated as a simple two-dimensional turbulent boundary layer, with the additional assumption that the meridional velocity component characterized the flow.

The annulus blockage due to the displacement thickness, δ^*, is computed by the following steps:

1. For a given meridional velocity distribution along the annulus wall, compute the momentum loss thickness, θ, from the equation:

$$\theta(m) = \theta_o + 0.006 \frac{\left[\int_o^m \{V_m(m)\}^4 \, dm\right]^{0.8}}{\{V_m(m)\}^{3.2}} \tag{68}$$

where θ_o is the inlet momentum thickness.

2. The shape factor at each station is then computed from:

$$H = 1.5 + 30 \left[\frac{d\theta}{dm}\right] \tag{69}$$

and the displacement thickness is given by:

$$\delta^* = (\theta)(H) \tag{70}$$

Jansen attributes Eq. (69) to Jules Dussord.

The procedure in applying the computation for displacement thickness (which is converted into a blockage at each station) is straightforward. At the conclusion of each major iteration as previously described the computer has available to it an array of values of meridional velocity along the inner and outer endwalls. Eqs. (68) and (70) are applied with an imposed value of inlet momentum thickness. The computed values of displacement thickness are converted into a blockage at each station. The next major iteration is then begun.

The procedure is a simple one; its justification lies in the fact that it appears to work. We give some examples below taken from Ref. 37; more follow in the next section.

Jansen had available to him when Ref. 37 was prepared both performance data and a geometrical definition of the Rolls-Royce Spey and Avon compressors, and he was given permission to use the information. Figs. 8 and 9 show pressure ratio and efficiency versus flow as calculated and as tested. Fig. 10 shows the computed blockage at two flows at 7008 R.P.M. The blockage reaches a value of 4.0% near stall.

Figs. 11 to 13 show similar comparisons for the 12-stage Spey compressor. Of particular interest is the value attained by the computed blockage. At 50.25 lbs/sec (at 10,674 RPM) the computed blockage is over 20%; at actual surge the value would be about 18%.

It is to be recognized that an identical computing routine was used for both compressors. The only difference was the input compressor geometry.

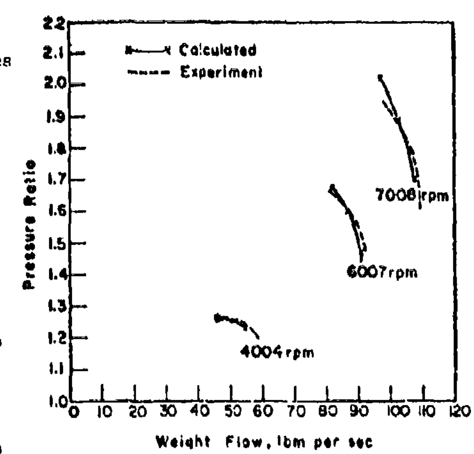

Figure 8

Avon compressor: comparison of calculated and experimental compressor performance.

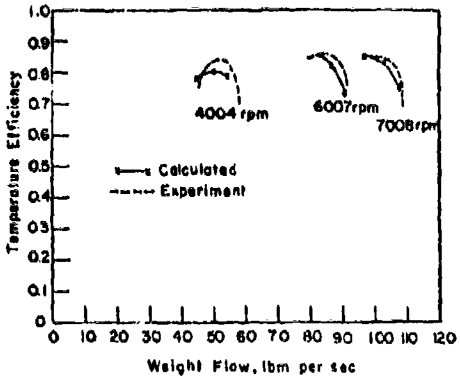

Figure 9

Avon compressor: comparison of calculated and experimental temperature efficiency.

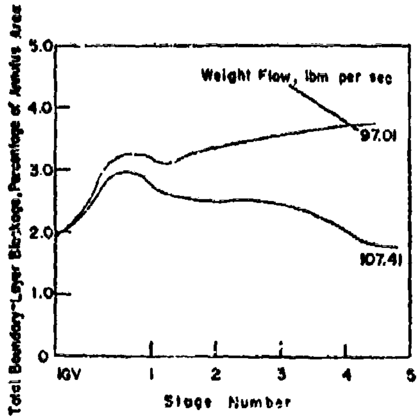

Figure 10

Avon compressor: boundary-layer blockage at 7008 rpm.

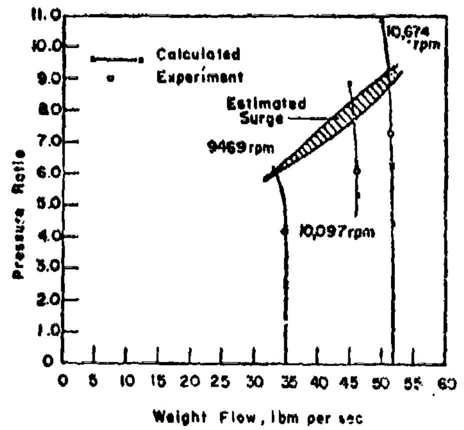

Figure 11

Spey compressor: comparison of calculated and experimental compressor performance.

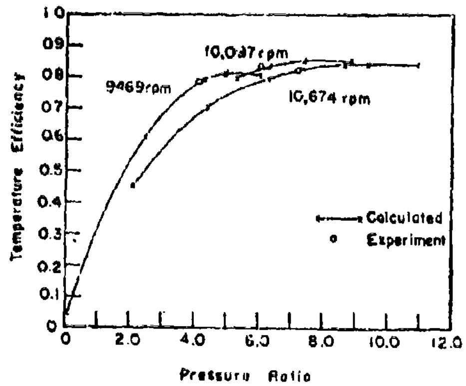

Figure 12

Spey compressor: comparison of calculated and experimental temperature efficiency.

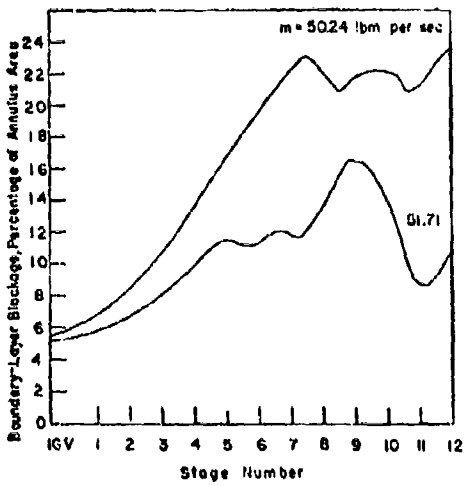

Figure 13

Spey compressor: boundary-layer blockage at 10,674 rpm.

ADDITIONAL EXAMPLES

The problem of securing convincing examples for an overall discussion of axial compressor computing procedures has always been a difficult one. Most data on multistage machines are proprietary. When it is possible to obtain overall or interstage data in the open literature, they are never accompanied with the necessary geometrical definition of the machine. There is, in fact, little but the NASA data of any value for a presentation such as this.

We will disucss three specific machines:

1. The NACA 5-Stage Transonic Compressor. Its design and experimental performance are given in Refs. 38 and 39.

2. The NACA 8-Stage Compressor. Its design and experimental performance are given in Refs. 40 and 41.

3. The GE/NASA High Mach Number 2D-Rotor. Its design and experimental performance are given in Refs. 42 and 43.

The three cases below were computed with a program basically similar to that of Ref. 37. It had, however, a totally different cascade subroutine (albeit based on the same data); most particularly the program is about twice as efficient in terms of computer time.

The NACA 5-Stage Transonic Compressor

Fig. 14 shows the overall performance (computed vs. experimental) which obtained with the NACA 5-Stage Transonic Compressor. The very foreshortened experimental curves resulted from the fact that this compressor was run as an engine; variations in flow, or pressure ratio, at any one speed could only be realized by engine exhaust nozzle area variation.

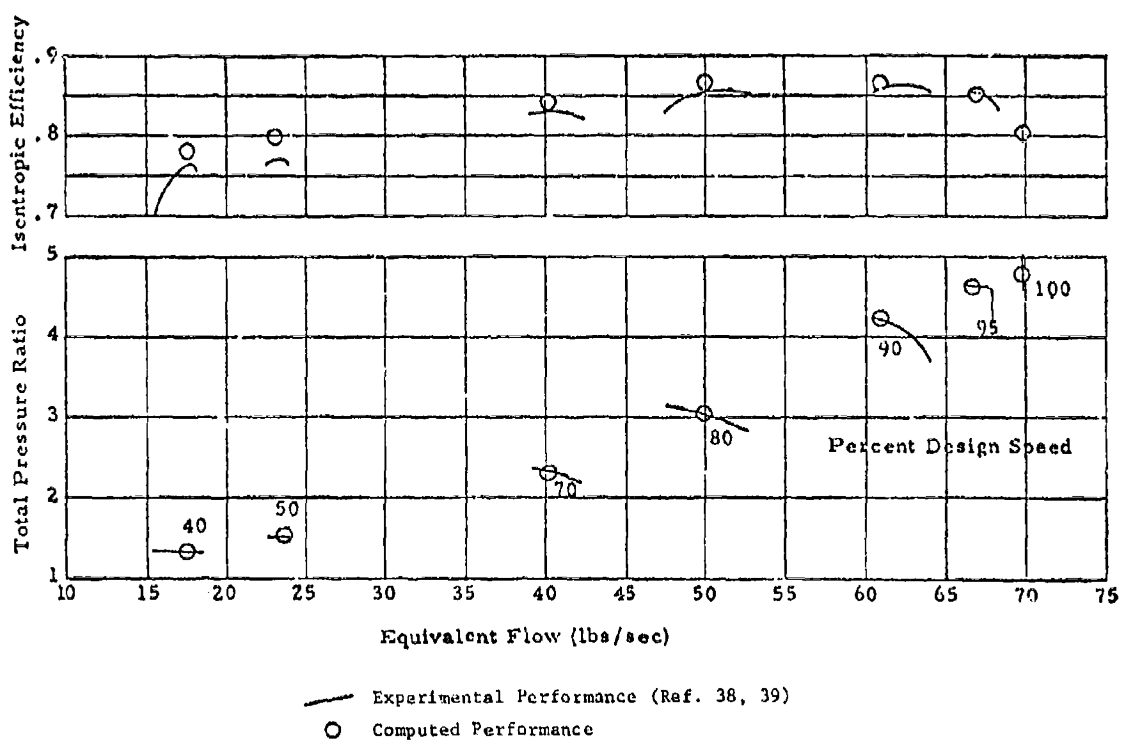

Figure 14

Overall Performance of NACA 5 Stage Compressor

The NACA 8-Stage Compressor

Figs. 15 and 16 show, respectively, the experimental vs. calculated comparison for flow vs. pressure ratio and flow vs. efficiency. The comparison is obviously an impressive one. It is to be stressed that the basic computer program - including every subroutine - is unchanged from that of the previous example. Only the input data - compressor geometry, flow and speed - are different between the two examples.

Fig. 17 shows the computed values of the blockage fraction for three points at 90% speed for the NACA 8-Stage machine. These are values which arise from a momentum integral solution basically identical to that described in the previous section.

GE/NASA High Mach Number 2D-Rotor

The two cases presented above show in a comparatively convincing fashion that the computer program is capable of giving creditable predictions of multistage machines. It has, in fact, been used on a large number of other multistage compressors of an industrial type with equally good results.

It is well, however, to be candid on this point. There are very few industrial axial compressors for which the test data can be trusted as being reliable. It is difficult to be certain whether the computing system is reliable or whether chance did not play a part in the excellent experimental vs. computed correspondences which has been realized.

It was decided, therefore, to test the system against a modern, high Mach number compressor for which reliable data appeared available, for which the geometry was well defined, and for which a considerable amount of radial traverse information was given in the report describing the test program.

The GE/NASA 2D-Rotor satisfied the requirements of a check in many ways:

1. It was a test of a single rotor, so that one could presume that the chances were maximized for reliable radial traverse data.

2. It - as contrasted with the other stages of this GE/NASA Contractor series - used a rotor with DCA-sections, for which the computer program was designed.

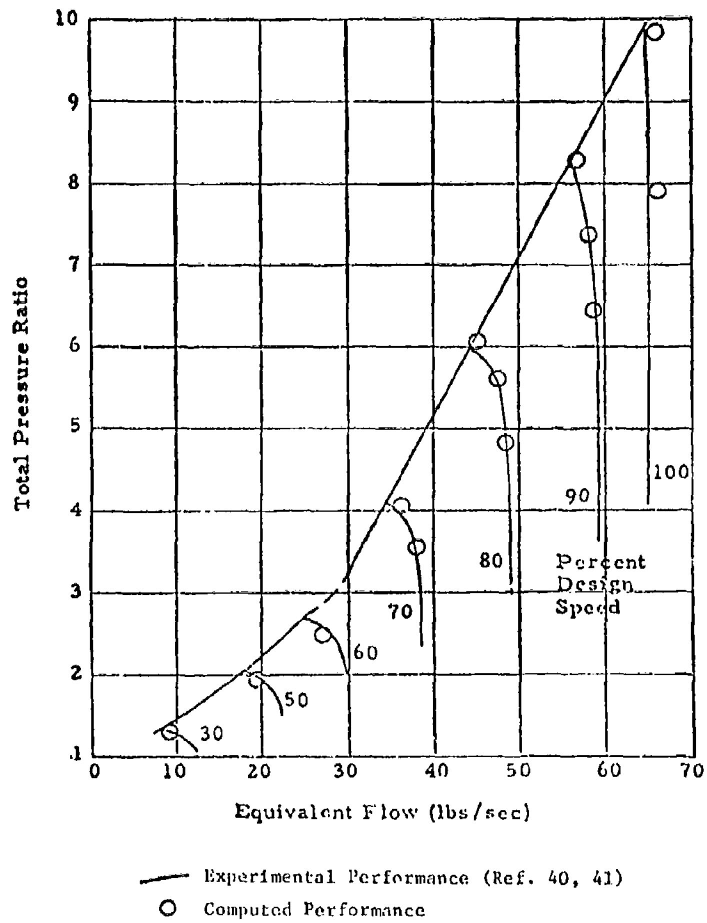

Figure 15

Overall Performance of NACA 8-Stage Compressor

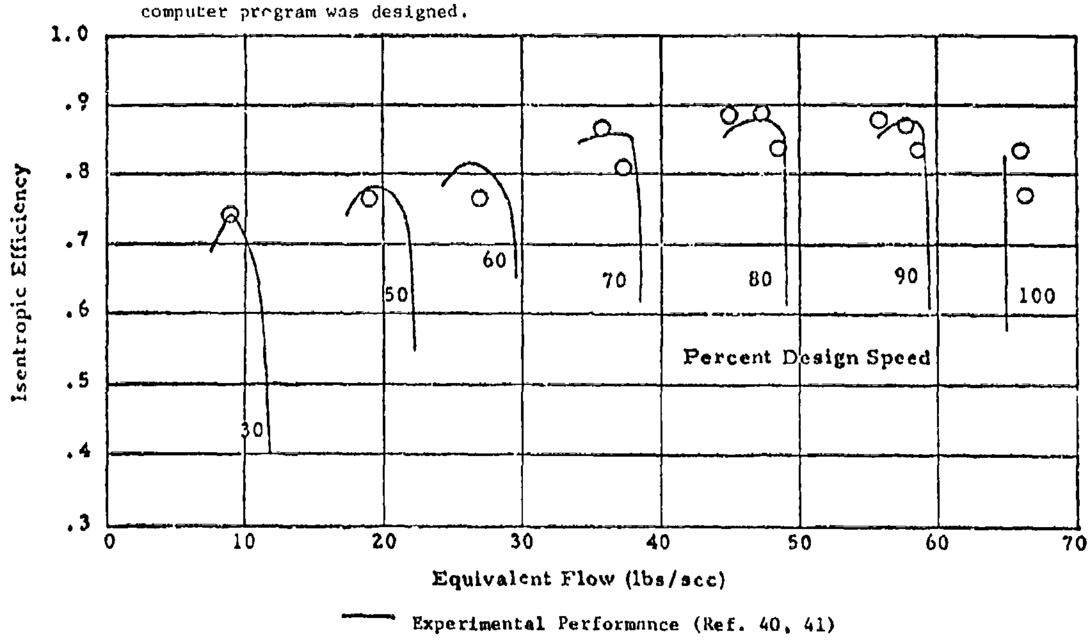

Figure 16

Overall Performance of NACA 8-Stage Compressor

3. The 100% speed design Mach number of the rotor was about 1.5; the 110% speed run carried the Mach number to 1.7. Inasmuch as the cascade option of the computer program was based primarily on NASA SP-36 data which did not extend much beyond a Mach number of 1.15 - 1.2, it would be of great interest to ascertain whether the analytic formulation - and its implied extrapolation built into the cascade subroutine - of the SP-36 data remained valid in the higher Mach number range.

4. Finally, the report on the stage contained geometrical data which were well-defined and interpretable.

It is to be noted that exactly the same computing routine, and the same options (i.e., additional tip loss, non-use of the supercritical deviation angle correction, as discussed above, etc.) were employed as were employed with the NACA 5-Stage and 8-Stage machines.

Fig. 18 shows the overall computed points - from 50% to 110% design speed - superimposed on the test curve given in Ref. 43. It is evident that the computing program is able to recognize what it is being asked to do.

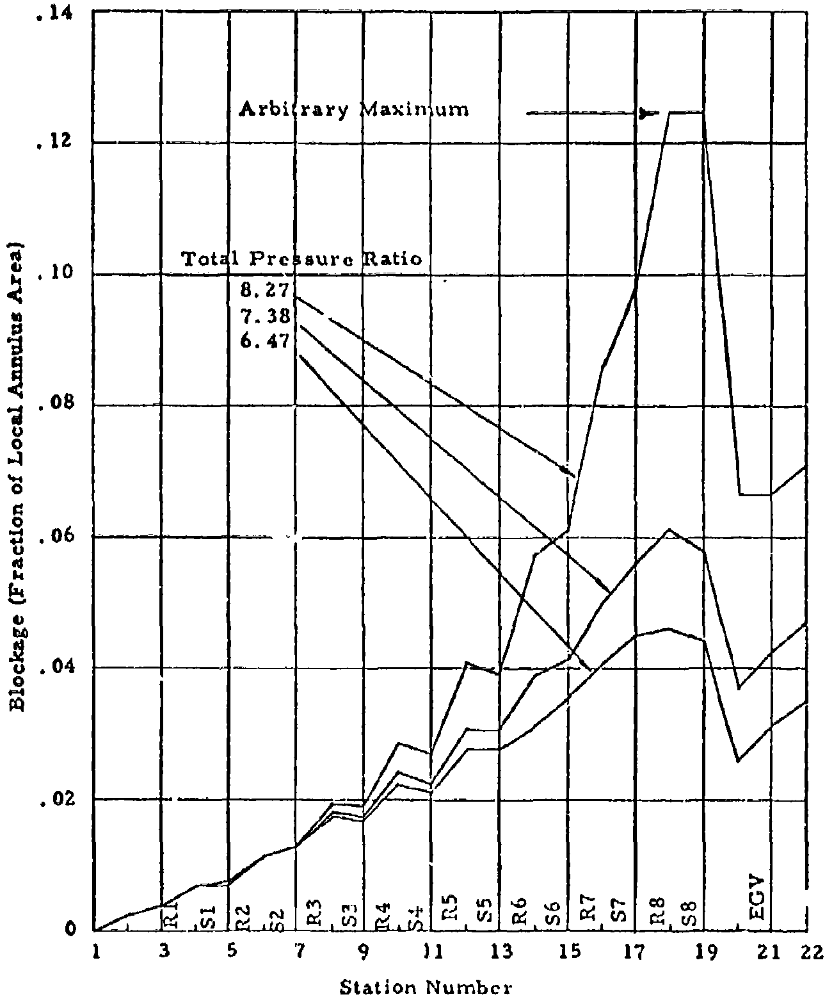

Figure 17

Computed Blockage Distributions for NACA 8-Stage Compressor at 90 Percent Speed.

One of the computing stations chosen at the rotor discharge was the plane at which was located the radial traverse probes (computed data could, therefore, give a direct comparison with the quantities measured, i.e., total pressure, total temperature and absolute angle). It is on the basis of these three numbers, along with mass flow and speed, that all other numbers were computed by G.E. in its preparation of the report.

Fig. 19 shows the comparison of computed vs. experimental temperature ratio for one point each at the five speeds ranging from 50% to 110%. Fig. 20 shows a similar comparison for pressure ratio; Fig. 21 for absolute rotor leaving angles.

A careful check of the data of Ref. 43 makes it evident that the computed results fall well within the experimental spread of the many test points taken. It is to be noted that the scatter of the test points at mid-span at high speed is undoubtedly due to the presence of the part-span shroud.

AXIAL COMPRESSOR DESIGN

It is not intended, in this final section, to discuss the design of axial compressors - except briefly in the context of what has already been stated. Some summation, however, appears to be in order, and it can be expressed in the context of the design process.

1. The chief emphasis of the Lecture has been on the analysis, or predictive, procedure. The reason for this is simple. If one can predict, with some assurance, the performance of a machine of arbitrary geometry, one has the most powerful design tool available.

2. The computer tools we have been discussing do, of course, have provisions for the normal design approaches. One can ask for vector diagrams consistent with a desired stage pressure ratio or temperature ratio, and answers will be forthcoming which completely satisfy the momentum/continuity conditions for an axisymmetric system. The problem is that one does not know, a priori, what the losses will be, especially since at an early stage of the design process the choice of blading has not yet been made.

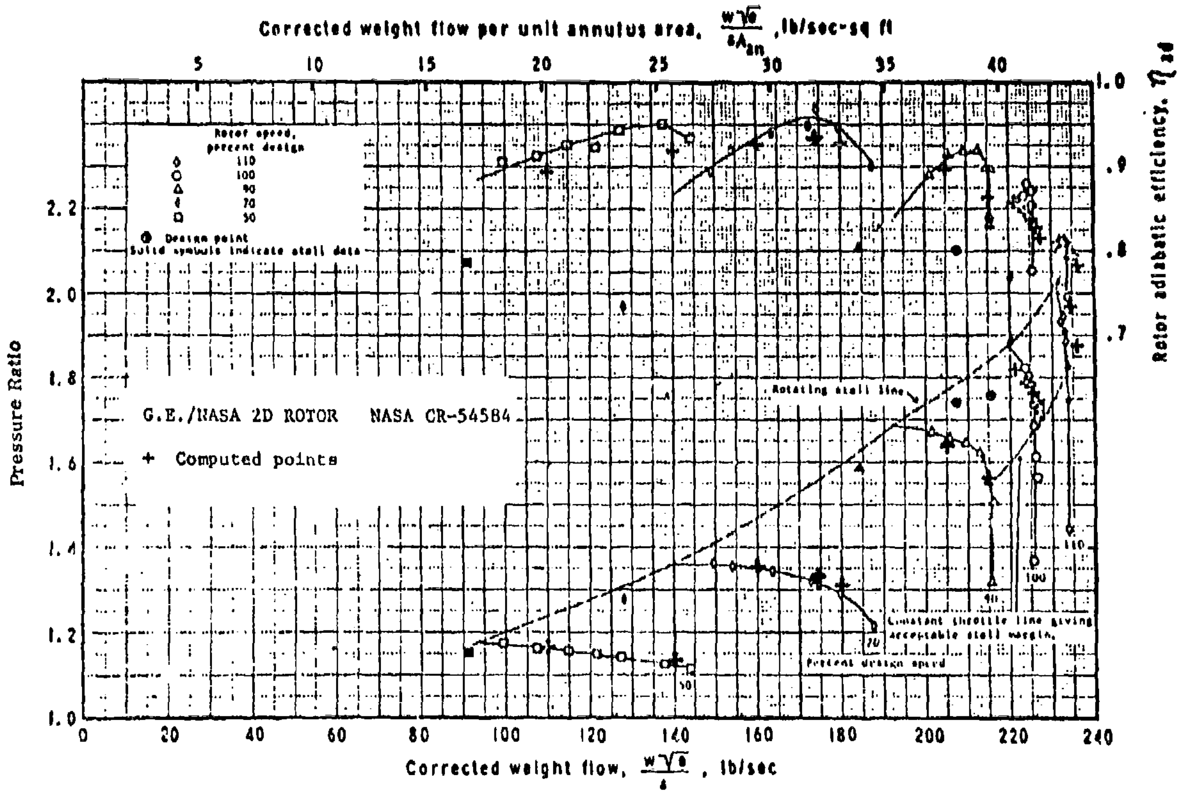

Figure 18

3. The valid answer to the dilemma, it is believed, is to use the computer program in the analysis mode for the design process. One performs a tentative design. The implied hardware is then entered into the analysis process. The answers, for a properly designed computer tool, give not only the performance of the initial geometry, but also the numbers which indicate where the tentative design is deficient. All incidence angles are printed out, along with a series of loss increments which permit a rapid assessment of what can be gained by changes - paper changes - to the initially prescribed hardware.

4. One proceeds, in short, as though the design process were built around a computerized experimental program, where blade leading and trailing edge angles are modified, solidity is varied and the annulus is changed until some satisfactory "optimum" is reached. The validity of the process depends upon the validity of the computerized predictive routine.

THE AXIAL TURBINE

The axial turbine, in principle, could be dealt with in much the same fashion as described for the axial compressor. In fact, however, it presents a significantly more difficult problem. One reason for this - a reason which is historical rather than fundamental - lies with the unavailability (in the public domain, at least) of turbine cascade data in a systematic form. A second reason arises from the strong secondary flow field which is generated in the high turning annular cascade of the turbine blade or vane row. This factor is particularly significant in the high radius ratio early stages of a multistage turbine. The endwall losses, and the cross channel and spanwise migration of low momentum fluid, can completely dominate the radial loss distribution at the blade row exit. For a heavily loaded early turbine stage, the "secondary" losses can easily outweigh in magnitude the profile losses which might be available from two-dimensional cascade testing. Closely related to this is the significantly greater effect of blade end clearance in the turbine as compared with the compressor.

There is strong evidence that even moderately loaded conventional turbine blade rows operate regularly with separation. Fig. 22, reproduced from Ref. 44 by Ainley and Mathieson, is a plot of profile loss coefficient versus Reynolds number for some impulse blade sections (upper group) and some reaction sections (lower group). Superimposed on the figure are two dotted lines representing the loss-Re slope (on a log-log plot) which one would expect to see with a laminar boundary layer (i.e., loss coefficient proportional to $(Re)^{-1/2}$), or that which would pertain for a turbulent boundary layer ($Y \approx (Re)^{-1/5}$).

It is evident that a number of the sections exhibit a slope greater than that to be expected either with a laminar or turbulent layer. This is a clear signal that some portion of the profile (probably the suction surface near the trailing edge) is separated, and that what is being measured is some degree of form drag.

At least two salient conclusions can be drawn from the suggestion implicit in Fig. 22:

1. Many turbine blade sections operate with partial separation, or at a condition perilously close to separation. Secondary and clearance flows, therefore, can readily trigger full separation.

2. The separated regions along the blade span provide spanwise paths along which the low momentum endwall flow can be transported. A clear case in point is the often noted pile-up of high loss at the inner radius of the turbine. The losses do not originate there; low momentum fluid has been transported there.

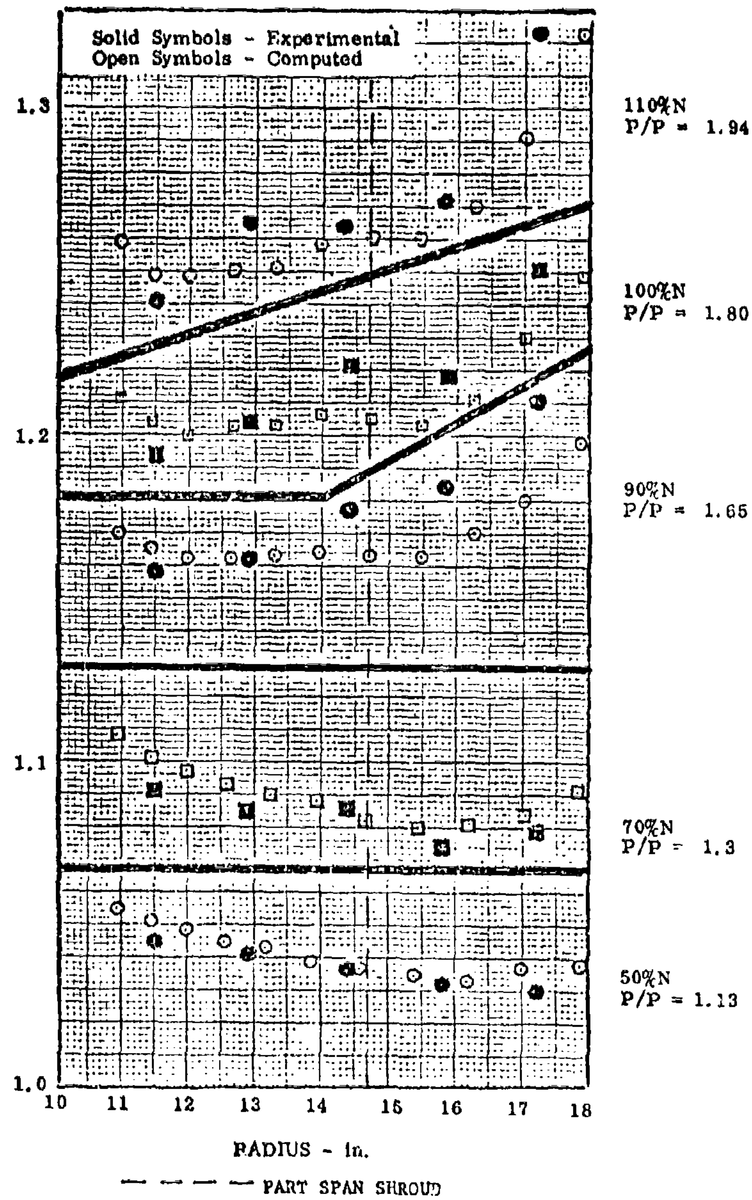

Figure 19

Temperature Ratio Comparison - GE/NASA Rotor 2D

Even in the highly accelerating flow field of the nozzle, separation is a danger. The rapid acceleration will, itself, re-laminarize the suction surface and endwall boundary layers (regardless of the upstream turbulence level). Under these circumstances, even a minimal disturbance or decellaration at or downstream of the throat can cause separation.

In the comparatively highly loaded axial compressor, the spanwise stream filaments maintain a near-identity, so that two-dimensional cascade data are largely applicable to the real stage; such is not generally the case for the turbine, especially in the early, low radius ratio early stages. In this regard we are forced to deal with a problem more analogous to the centrifugal compressor, where nothing equivalent to compressor cascade data exist or are relevant. In short, the basic flow field is often truly three-dimensional, and it is not readily dealt with by the two-dimensional methods which can be used for axial compressors. One is better advised to fall back on the one-dimensional technique of Ref. .

The final problem in predicting the performance of an axial turbine is very simply - a computational one. If the turbine were of a lightly-loaded, high aspect ratio configuration, a computing technique very similar to that described for axial compressors would be quite applicable when the turbine operating back pressure is high enough. As the flow is increased at a fixed rotation speed, a point will be reached where choke will occur in some blade row, probably in the first nozzle.

The choke occurs in the blade row throat; in order to construct a realistic computing routine, therefore, throat stations within each blade row must be included, in addition to stations at the blade leading and trailing edge. At the throat stations, angles and blade thickness must be specified as a function of span. This, in itself, is readily done. The real problem is that, at the throat station, one must have the solution corresponding to the maximum flow as discussed earlier. Computationally (as has already been stressed) this is quite difficult to obtain. There is the additional complexity that choke does not occur simultaneously across the span. At full choke, some sections can still be subsonic and others supersonic.

The requirement for a lower overall turbine back pressure can only be met by changing the choked cascade downstream angle and loss, but only for those sections to the right of the peak plot. This ~~... ...ated flow two-dimensional cascade data~~, or a subsidiary computation equivalent in some sense to

a method of characteristics solution for the region downstream of the throat. In any event, the choked cascade downstream angles are modified (in the direction of a decrease static pressure and density), and the computation can be continued until choke is reached at some downstream blade row, or until the downstream back pressure is attained.

It is believed that the complete process described (sketchily) above has not yet been implemented. The immediate answer to the multistage axial turbine off-design prediction problem seems still to be some version of Ref. 44.

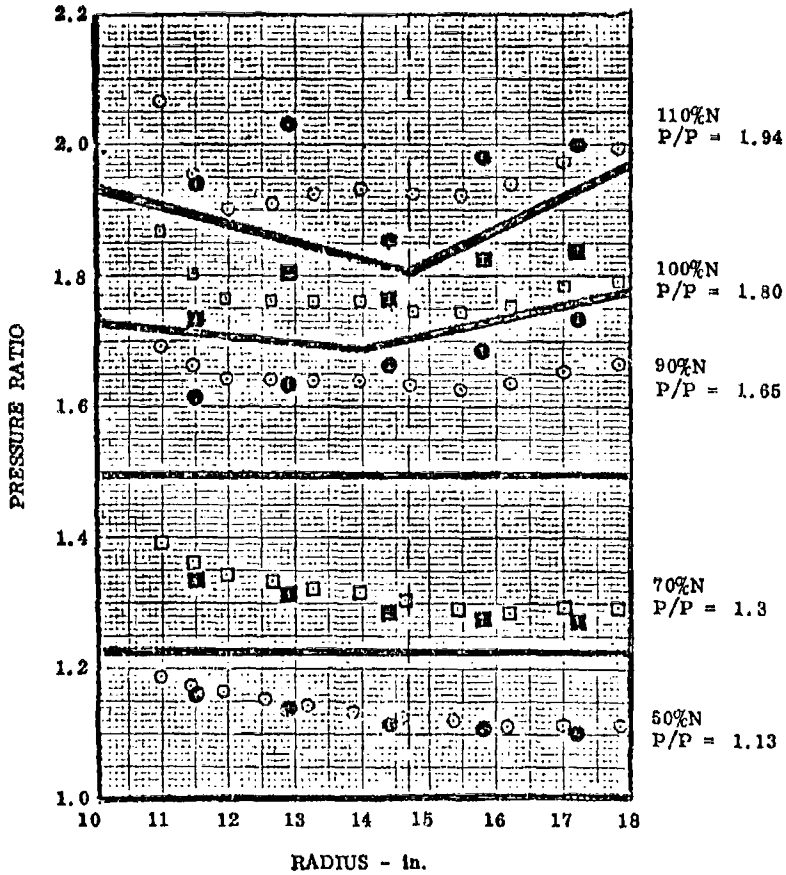

Figure 20

Pressure Ratio Comparison - GE/NASA Rotor 2D

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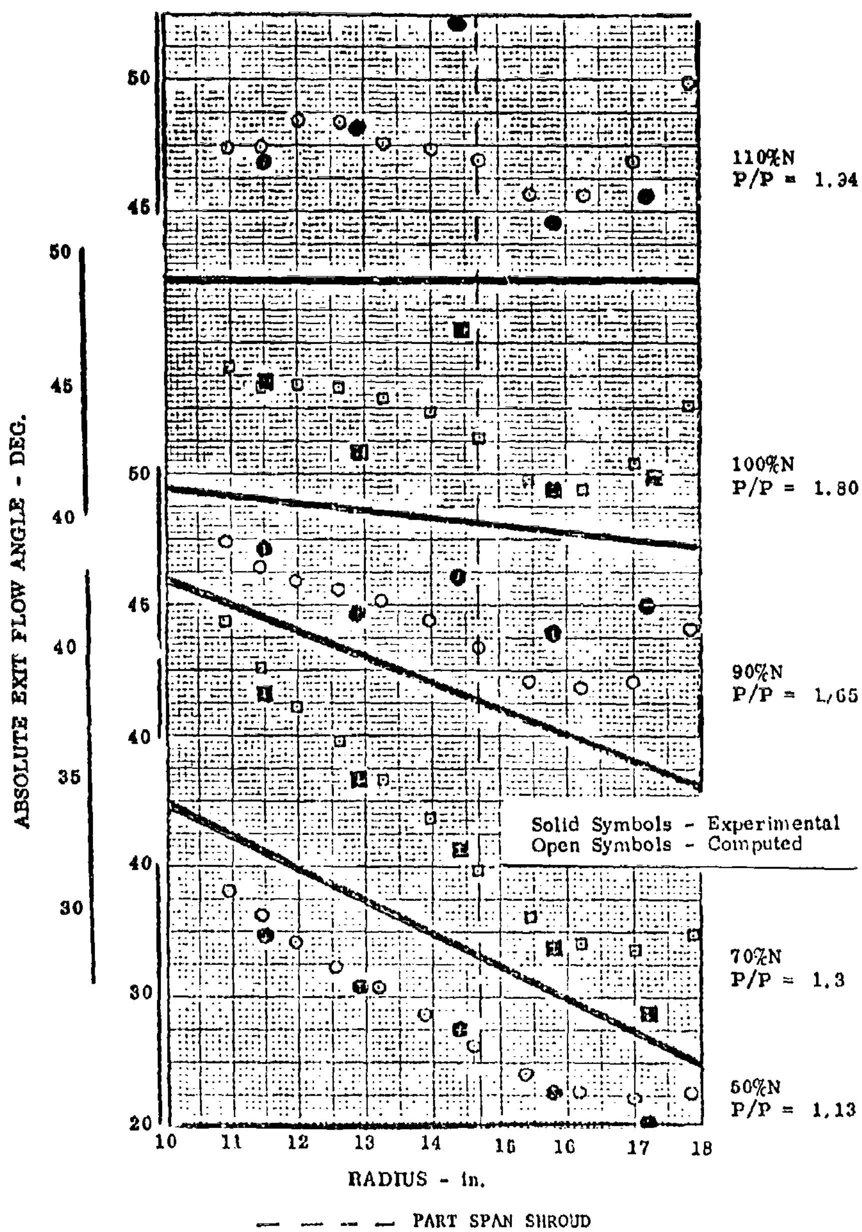

Figure 21

Absolute Exit Flow Angle Comparison GE/NASA Rotor 2D

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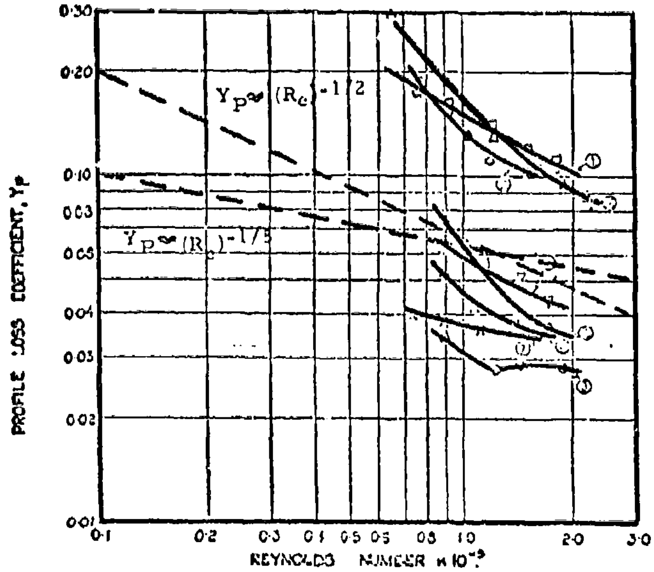

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DESIGN OPTIMIZATION AND PERFORMANCE MAP PREDICTION FOR CENTRIFUGAL COMPRESSORS AND RADIAL INFLOW TURBINES

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SUMMARY

The initial specification of compressor and turbine geometry and performance characteristics, including operating maps, can follow different paths depending on the degree of departure from previous design experience. Principal attention is focused on totally new design problems requiring systematic design optimization to meet performance criteria under diverse operating conditions. The fundamental flow physics involved for both the centrifugal compressor and radial inflow turbine are briefly reviewed with principal attention focused on the strategy used for selecting optimum stage configurations. The performance map is obtained from the final step of this design optimization exercise.

1. INTRODUCTION

During the past two decades, radial turbomachinery has come to play a very strong role in the production, distribution and conservation of energy. Radial turbomachinery applications include process compressors for the production of LNG and various feedstocks, aircraft gas turbines, automotive (truck and automobile) gas turbines, diesel (from 200 to 40,000 hp) turbochargers, refrigeration, cryogenic compression and expansion plus a variety of new turbine expander applications for waste heat recovery. Impeller sizes vary from the diameter of one's little finger to one's height. Efficiency levels have improved by as much as 10 points over the past two decades in certain application areas. Radial turbomachinery has replaced other types of equipment in many of the applications mentioned. In spite of improvements made in various systems, there is intense pressure today in all of these areas to further improve performance levels. The intense efforts by various designers and researchers today at least atest to the desire for improved performance and the belief that improvements are possible.

Economic leverage for improved turbomachinery performance is substantial. Typical examples range from about $300 in yearly operation expense for a single point of improvement in a 1000 ton refrigeration compressor, to $500,000 for a single point of improvement in hydraulic turbines for electric power production, see Japikse (1975). Ecological demands for reduced machinery noise has placed further attention on turbomachinery as opposed to reciprocating equipment.

Prospects for using small gas turbines, which depend heavily on high performance radial stages, are excellent. Perhaps the best known application is the helicopter gas turbine engine which frequently employs radial stages. Further, the automotive gas turbine is structured around a centrifugal compressor. A number of small, intriguing designs have evolved over the past several years ranging from 10 hp to 1,000 hp using a single radial compressor stage and either a single radial turbine stage with direct shaft power output or with a radial gasifier turbine stage and an axial power turbine stage. A number of such units are now available on the market; a variety of others will soon be available ranging from snowmobile engines to drone engines.

Invariably the advanced designer is involved in a search for optimum performance elements and optimum performance systems. To be sure, there is a considerable margin for improvement in centrifugal compressors and in radial inflow turbines. Dean (1974) has indicated a margin of 5 to 10 points through which we can expect the centrifugal compressor to grow during the next two decades. Quite likely, there is a similar margin for radial inflow turbines. Nonetheless, we are still far removed from these performance levels in most industrial machinery today.

This paper is one of a series of lectures sponsored by AGARD on the matching and performance prediction of gas turbine systems. In this paper, a design methodology developed to optimize radial turbomachinery is presented. The emphasis here is on the design approach taken to yield a high performance gas turbine system. A 25 SHP gas turbine will be considered as a preliminary design sample study. The basic technology upon which this design is based is detailed in various other reports (for example: Dean (1974), Runstadler et al. (1975), Rothe and Japikse (1975), and Japikse et al. (1975)). Many individuals at Creare have contributed to the evolution of the design approach which is detailed herein.

2. APPROACH TO SYSTEM MATCHING

The design of an outstanding piece of machinery must begin with a very specific concept, hopefully an inspired synergy of various high performance components, which makes the new design unique in various marketable fashions. The designs and technology base for the various components may come from diverse sources; but the overall specific concept must be evolved, developed and championed by an individual systems engineer or a

system group. The system specialist is responsible for the overall synthesis of the unique aspects of each of the different system components. The system must meet overall performance requirements (thrust or power level, fuel rate, inertia, etc.), cost, size, weight, and life. The systems engineer must trade these factors to find an optimum for the market place.

To guide initial engine system planning, the systems specialist must have a large collection of general data at his disposal to describe the performance of each component as a function of the major parameters which influence cost, size and life. An understanding must also be reached concerning the use of current technology or whether advanced technology should be pursued with the concomitant cost of developing advanced performance components. Therefore, a very accurate understanding of the data base from which the system is configured must be at hand.

To illustrate this process, and to set the stage for the preliminary design of several radial turbomachinery stages in subsequent sections of this paper, we will consider as a design example a 25 shaft horsepower (SHP) small gas turbine (SGT). No effort will be made here to systematically optimize all parameters which influence a good system selection (although this is needed in actual practice); rather sensible values of the parameters will be arbitrarily selected to illustrate the key points which must be considered before an overall system is defined. With this background, we can then examine the detailed problems which confront the aerodynamicist during radial compressor and turbine design optimization.

Life is probably the first consideration which a designer must face. Frequently, one will push against the limits of component life in choosing aerodynamic parameters. For our 25 SHP example, we will assume that the compressor can operate at a tip speed of approximately 1700 ft/sec. In fact low cost, long life impellers of die cast or investment cast aluminum alloys usually are restricted to 1500 ft/sec or below. Moderate life components manufactured by the same approach can tolerate speeds in the range of 1600-1700 ft/sec. Machined impellers of aluminum or titanium can tolerate speeds of 2,000-2,200 ft/sec whereas speeds as high as 2,500 ft/sec have been attempted for modest life using impellers machined out of forged titanium blocks. Similar restrictions are felt on the turbine end and here the turbine inlet temperature becomes a controlling parameter. If the turbine inlet temperature is held at 1500°F or below, then a variety of comparatively inexpensive high temperature alloys are available from which uncooled rotors can be fabricated. An introduction to high temperature design problems and life calculation techniques has been summarized by Japikse (1976).

With the compressor tip speed set at 1700 ft/sec, we find the compressor specific work directly from the Euler turbine pump equation:

$$W_{xc} = \mu \cdot u_2^2/gJ \qquad \text{(no preswirl)} \tag{1}$$

The work input coefficient, μ, can vary between 0.7 and 0.9 depending on the amount of blade backsweep, the number of blades and a variety of lesser parameters. Assuming that broad range is required for the compressor, a backsweep angle on the order of 30° or 40° may be desirable and a low work input coefficient would result for this preliminary design evaluation. With the specific work input set, the pressure ratio of a compressor is nearly defined. In fact, working from the definition of compressor efficiency, we can estimate the stage pressure ratio as follows:

$$pr = (W_{xc}\eta_c/C_pT_{00} + 1)^{k/k-1}. \tag{2}$$

For a tip speed of 1700 ft/sec, W_{xc} = 80.75 Btu/lbm and pr = 3.87. To carry out this calculation, $\eta_{TS,c}$ = 0.757 was assumed (this will be explained later).

Following through the gas turbine configuration, the compressor discharge air enters a combustion chamber and then proceeds through the turbine. A regenerative or recuperative cycle might be employed which adds another loss producing element between the compressor discharge and turbine inlet. For the present example, we assume that the turbine inlet pressure is equal to 0.92 times the compressor discharge pressure. Furthermore, we will consider only a single shaft configuration where the load and compressor are driven together. Turbine inlet temperature equal to 1500°F will be taken. Thus the system restraints now have specified the turbine pressure ratio and the definition of turbine efficiency can be used to determine the turbine specific work as follows:

$$W_{xt} = \eta_t c_p T_{03}((er)^{k-1/k} - 1) \qquad er = \chi pr,\ \chi = 0.92 \tag{3}$$

from which we directly compute that the turbine specific work is 129.8 Btu/lbm. This calculation requires an iteration to define the turbine exhaust temperature. A turbine efficiency of 0.825 (TS, adiabatic) was assumed (to be explained later). In turn, the tip speed can be computed to be 1767 ft/sec (via $W_{xt} = u_2^2/gJ$, for zero exit swirl).

With these conditions set, we can determine the gas turbine mass flow knowing the required shaft power output. We find:

$$P_s + P_c + P_b = P_t, \quad P_b = (1-X)P_t \tag{4}$$

$$P_s + mW_{xc} = mXW_{xt}(1 + m_f/m) \tag{5}$$

$$\dot{m} = P_s/(X(1 + m_f/m)W_{xt} - W_{xc}). \tag{6}$$

Assuming that 8% of the turbine power is required by the bearing system, and a nominal fuel rate, we compute that the mass flow for this proposed system must be 0.475 lbm/sec.

Finally, it is necessary to consider shaft speed. As long as the speed is reasonable (i.e. does not imply a short life or a high cost), the system engineer does not care what the speed is. However, it is important to the aerodynamicist. In a later section, we will see the role specific speed plays in setting stage performance. A variety of possible speeds for this sample case are shown in Table 1 below.

Table 1 Speed/Radius Alternatives For the 25 SHP SGT

N	N_{ssc}	N_{sst}	$r_{t,2}$
50,000	39.6	54.8	4.05
75,000	59.4	82.2	2.70
100,000	79.2	109.6	2.02
125,000	99.0	137.0	1.62

These few equations and comments serve as an introductory example of the type of iterations which are necessary to come to an initial cycle specification. Many cycle optimization schemes can be evaluated; some tip speeds lead to better component life and higher stage efficiency - but at the expense of size. Further, the efficiency levels assumed above must be consistent with the speed selected from Table 1 which in turn will be influenced by the component diameter limit, a frequent design restraint!

To assure a successful design project, it is necessary that the aerodynamic designer supply simple but accurate performance data, subject to all parameters which are of significance vis-à-vis cost and life to the system specialist. Invariably, this requires some representation of present state-of-the-art performance.

3. STATE-OF-THE-ART RADIAL FLOW COMPRESSOR/TURBINE PERFORMANCE

Past experience is invariably the first level of knowledge employed in the definition of any new gas turbine design. Data acquired from previous radial turbomachinery tests can be examined in terms of basic non-dimensional parameters to reveal overall trends. Isentropic specific speed (N_{ss}), stage pressure ratio or expansion ratio (pr or er) and Reynolds number (Re) as well as critical geometric scale parameters (such as tip clearance) can be appropriately used to record overall past experience.

The use of these parameters is based on dimensional analysis theory which is well known. However, these variables must be used with caution. If one has two machines which are geometrically similar and each operate at the same specific speed, then identical aerodynamic states will be found in each machine, if Reynolds number effects (viscous effects) are insignificant. Thus specific speed is a good parameter for scaling geometrically similar stages; however, it does not give us a simple and clear correlating parameter for all machines of different geometric types. Therefore, it is not surprising that a survey of radial turbomachinery state-of-the-art performance will employ a variety of parameters. In part, performance representation of current machines is confused by inadequate data reporting by various sources.

The level of efficiency achievable at various pressure ratios and flow levels is illustrated in Figure 1. Only those stages which show the highest performance at any given pressure ratio have been plotted. Data has been extracted from various sources. It must be recognized that there is considerable room for interpretation of the data shown. We prefer to work with adiabatic efficiency, yet this is rarely established in conventional tests reported in the open literature. The variation in performance with mass flow is due in part to Reynolds number effects which, at a given pressure ratio, would suggest that higher efficiency would be obtained at higher Reynolds numbers. Conventional Reynolds number scaling rules indicate that only half the mass flow variation (Figure 1) can be attributed to Reynolds number effects. Probably, a scale effect is also involved. At low flows, very narrow impeller tip depths are involved and even a minimum running clearance begins to amount to a significant tip clearance penalty. Thus the trends shown in Figure 1 are confused* by non-adiabatic conditions, by Reynolds

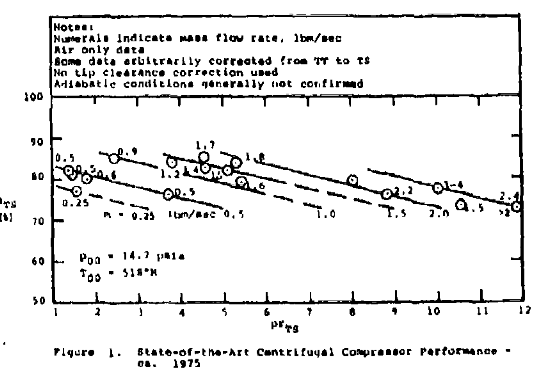

Figure 1. State-of-the-Art Centrifugal Compressor Performance - ca. 1975

* Presently, the data used to compile Figures 1-4 is being reviewed to obtain modified performance maps with tip clearance, heat loss, and Reynolds number effects removed.

ber effects and by tip clearance variations. Finally, the solid and dashed lines shown in Figure 1 indicate possible mass flow trends; but a careful examination of the data shown indicates that the lines could have been constructed with less slope if a few data points were given greater weight.

Unfortunately, a centrifugal compressor can rarely be designed with high efficiency as the only objective. Stable operating range is usually of equal importance. It is an unfortunate fact that design techniques used to extend range often come at the expense of efficiency. As a rule, design artifices introduced to extend range are characterized by stable (i.e. negatively sloped pressure ratio versus flow) characteristics which result from some loss producing flow effect (throttles, etc.). Figure 2 illustrates the degradation of efficiency as greater range is achieved for a variety of different pressure ratio ranges. Also shown is the reduced range which is achieved at very high pressure ratios. Perhaps of greater significance, is the exception to the range versus efficiency trade for stages in the pressurerange of 3 to 4; here, we see that a variety of machines suggest that greater range is achieved with increased efficiency (the truth is unclear).

Figure 3 parallels Figure 1, but for radial inflow turbines. As a rule, these turbines have reaction levels between 30% and 50%. Again, a clear degradation of efficiency with expansion ratio is evident. In this case, it is clear that viscous (Reynolds number) effects do not play a key role. Instead, it appears that tip clearance penalties are reduced at high flow levels where large tip widths are employed.

Finally, Figure 4 shows a collection of data plotted as turbine efficiency versus specific speed. It must be stressed that each point in this figure illustrates a completely distinct geometric configuration. If an individual configuration is tested over a variety of pressure and flow conditions, an η_{TS} vs. N_{SS} trend would be achieved, but this would not be the same curve shown in Figure 4. Furthermore, the best performance curve is not sacred. This curve will change during subsequent years as further attempts are made to design high performance stages at various specific speeds. The important point, however, is that there is a wide range of specific speed across which high performance stages can be designed (from N_{SS} = 50 to N_{SS} = 80, there is only a 2 point variation in efficiency). Incidentally, a similar plot can be established for the centrifugal compressor with an equivalent wide range in efficiency with specific speed.

Data of the type shown in Figures 1 - 4 suffice to give an early estimate of performance levels which might be reached in a new gas turbine design, such as the 25 SHP example considered herein. But it must be recognized that correlations of the type revealed in Figures 1 - 4 are fraught with uncertainties and serve to correlate ignorance as well as knowledge.

Returning to the sample matching problem considered in the previous section, we see that the expected efficiency (and range) levels must be carefully chosen as a function of pressure ratio, mass flow, and clearance effects. In the previous numerical example, this was done using the data of Figures 2 and 3. In fact, efficiency was

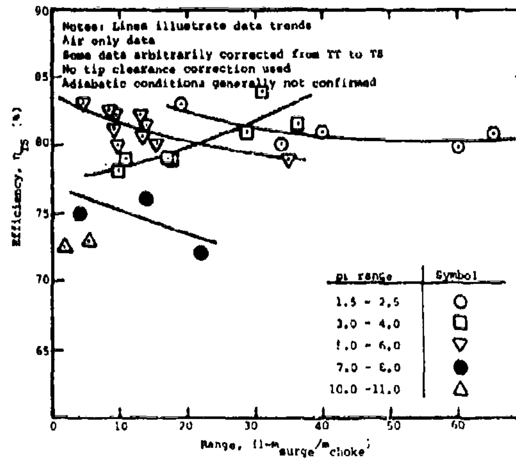

Figure 2. State-of-the-Art Centrifugal Compressor Efficiency Versus Range, ca. 1975.

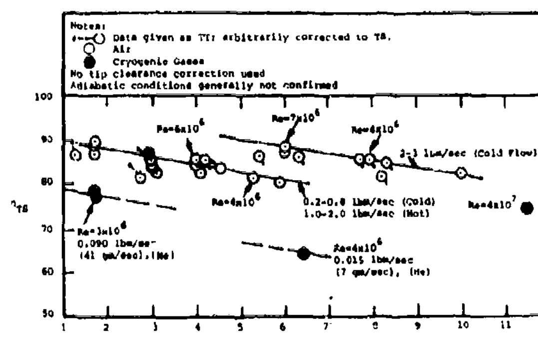

Figure 3. State-of-the-Art Radial Reaction Turbine Performance - ca 1975

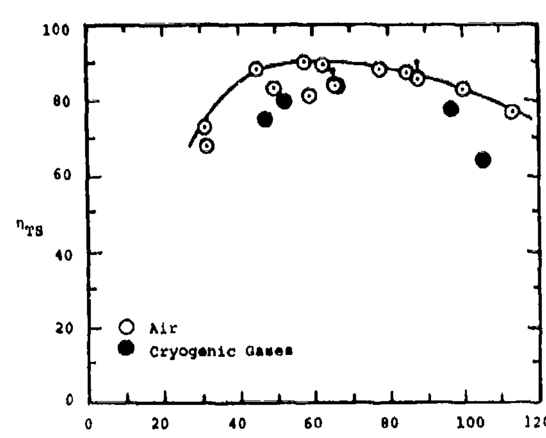

Figure 4. Efficiency vs. Specific Speed for Radial Inflow Reaction Turbines—State-of-the-Art, ca. 1975

$$\eta_{TS,c} = 0.8557 - 0.0257 \text{ pr} \quad (7)$$

$$\eta_{TS,t} = 0.8875 - 0.0175 \text{ er} \quad (8)$$

Data shown in Figure 4 (and equivalent data for centrifugal compressors) could used to choose a desired operating speed. However, while the η versus N_{ss} data provides a wide correlation of what has been achieved for a variety of machines, it does not answer two basic questions:

1. What is the variation of η_{TS} vs. N_{ss} for a specific application type. (for example our 25 SHP gas turbine), and
2. What is the optimum efficiency which can be achieved, given the particular design conditions (cost, size and life).

An example of computed efficiency vs. specific speed trends for three different applications of centrifugal compressors is shown in Figure 5. Clearly the specific speed for peak efficiency varies depending on the design situation. The information learned from the 50 SHP SGT study (Figure 5) can be used to guide the selection of shaft speed for the 25 SHP case. Using Figure 5 to represent the compressor, and Figure 4 to represent the turbine, we then find the following match possibilities:

Table 2 Speed Selection For Peak Efficiencies

N	$N_{ss,c}$	$N_{ss,t}$	$\eta_{TS,c}$,%	$\eta_{TS,t}$,%*
50,000	39.6	54.8	77	89
75,000	59.4	82.2	79	88
100,000	79.2	109.6	80	83
125,000	99.0	137.0	79	60(?)

* Data from Figure 4 may not meet item 2, above.

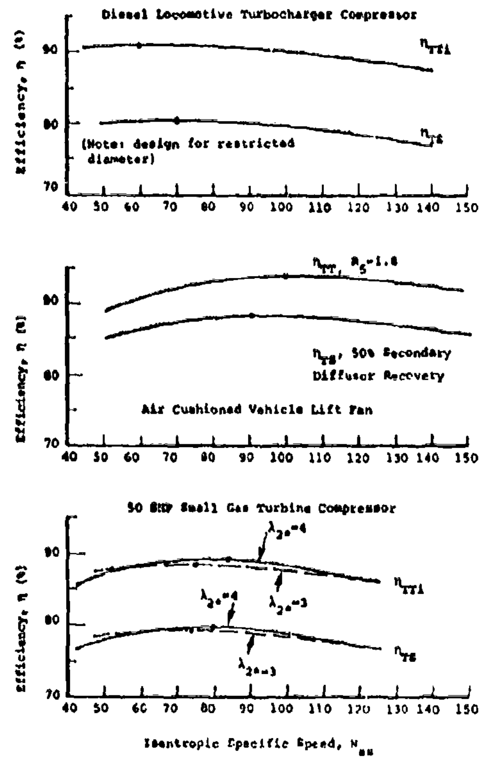

Figure 5. Computed $\eta_{TS,i}$ vs. N_{ss} Trends for Different Applications Showing Various Values (—+—) for Optimum Design N_{ss}.

With this information a speed of 75,000 rpm is choosen which implies an overall turbine diameter of approximately 9.5 inches. If this diameter is too large for the particular 25 SHP SGT problem, then it is necessary to go to a higher speed and to pay a concomitant efficiency penalty as shown in Table 2.

4. PRELIMINARY DESIGN AND PERFORMANCE OPTIMIZATION

Design Approach

Based on the previous matching and performance data, preliminary performance objectives are set as follows:

Table 3 Preliminary Performance Objectives for the 25 SHP SGT Example

Parameter	Compressor	Turbine
m	--0.475 lbm/sec--	
N	----75,000 rpm---	
pr,er	3.87	3.56
r_2	2.45in	2.70in
η(SOA)	75.7%	82.5%
range	20.0%	--

The above characteristics are a starting point. To effect an aerodynamically advanced design, it is necessary to evaluate every fundamental aerodynamic characteristic of the proposed centrifugal compressor and radial inflow turbine. If possible, we would like to push beyond state-of-the-art performance levels which lead to Table 3, above. Unfortunately, there are more than 20 distinct variables for the compressor and turbine

each. It is impossible to independently study the performance of each variable over the full range of possible combinations. Some designers resolve this dilemma, after a fashion, by simply using gross loss coefficients to correlate general performance characteristics on a component-by-component basis (impeller, overall diffuser, etc.). Unfortunately, such an approach does not offer much hope for systematically optimizing a stage for peak performance.

An alternative approach is to reduce each gas-dynamic performance characteristic of the centrifugal compressor and radial inflow turbine to its most fundamental aerodynamic phenomena and seek out the controlling variables for each process involved. For preliminary design and stage optimization, this is most conveniently done on a one-dimensional, station-by-station basis.

In our research and design experience, we have found that the critical aerodynamic phenomena in either the centrifugal compressor or in the radial turbine are the diffusion processes, not the loss generating processes per se. That is, high performance levels can be achieved best with an intimate knowledge of the basic diffusion processes; in turn, it is comparatively easy to establish levels of entropy production (total pressure loss). Details of the technology base which is employed for the optimum design of centrifugal compressors is detailed by Dean (1974), Runstadler et al. (1975), Rothe and Japikse (1975) and Japikse et al. (1975). In the following sections, we will illustrate the use of this technology base in a design optimization study for the 25 SHP SGT.

The first step in the detailed specification of a compressor or turbine stage involves certain optimum choices for the velocity triangles at the inlet and discharge of each stage component. For example, for any mass flow, speed, and hub-tip ratio, there is an optimum inlet axial velocity to achieve minimum (compressor) inducer tip relative Mach number. This effect is shown in Figure 6 where a tip relative Mach number of approximately 0.7 can be obtained for our particular compressor design with an inlet axial velocity of approximately 400 ft/sec. If manufacturing requirements had stipulated a higher hub-tip ratio, we would be forced to accept a higher tip relative Mach number. If we were to design at higher specific speed, we would have to accept higher tip relative mach numbers which means a higher relative kinetic energy and a much more difficult impeller performance design task. This effect is demonstrated by the parameter IKE which is the inlet relative kinetic energy divided by the work input. IKE is shown in Figure 9 plotted against specific speed. At higher specific speeds, IKE becomes more potent and achieving an efficient impeller is increasingly difficult. The selection of an optimum axial velocity (to minimize $M_{rel\ 1t}$) also implies the blade setting angle. In Figure 7 we have illustrated the variation in blade angle at different specific speeds and different inlet axial velocities. Here, we are not faced with any manufacturing or performance problem.

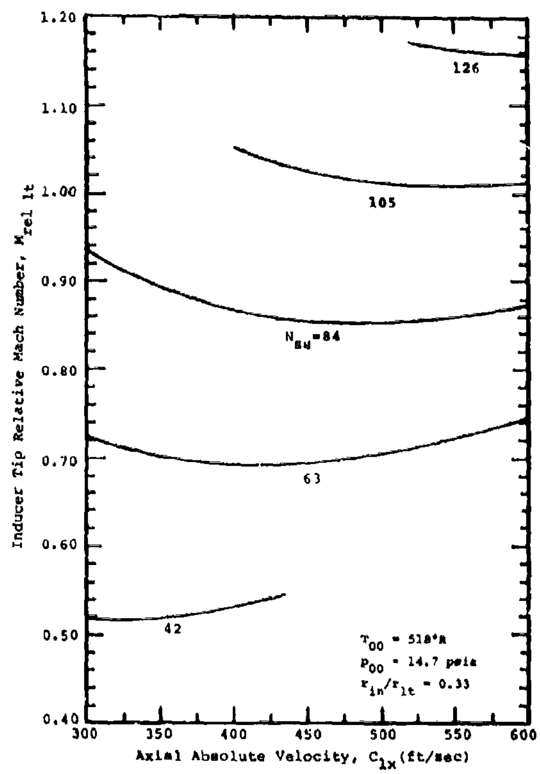

Figure 6. Inducer Tip Relative Mach Number Variations

The next station in the centrifugal compressor design is the impeller discharge/diffuser inlet. Here we must prescribe the exit swirl parameter and the blade backsweep angle to obtain the best stage performance within size restrictions (if any). We have established to our satisfaction that impellers can be designed with or without significant backsweep and still achieve excellent total-to-total efficiency and excellent impeller relative diffusion. The choice of exit swirl angle (or the tangent of the angle, λ_{2*}) and blade backsweep angle is made to facilitate good diffuser design (within size restriction).

The use of appreciable blade backsweep and moderately high exit swirl parameters (λ_{2*}) will contribute to improved range and improved diffuser performance. There are geometric restrictions. Increased blade backsweep implies a larger impeller diameter. Increased λ_{2*} implies very tight diffuser inlet angles and the design of a good diffuser leading edge may become difficult or impossible. Generally, 10° of blade backsweep is worth 1 to 2 points of stage efficiency and the usable flow range at 100% speed is frequently doubled over a simple radial discharge impeller for 30° - 40° backsweep. The blade backsweep angle, β_{2b}, is set equal to -40° for this study which corresponds roughly to the upper limit permitted by stress considerations. Figure 5 shows, for a 50 SHP SGT the impact of λ_{2*} on stage performance.

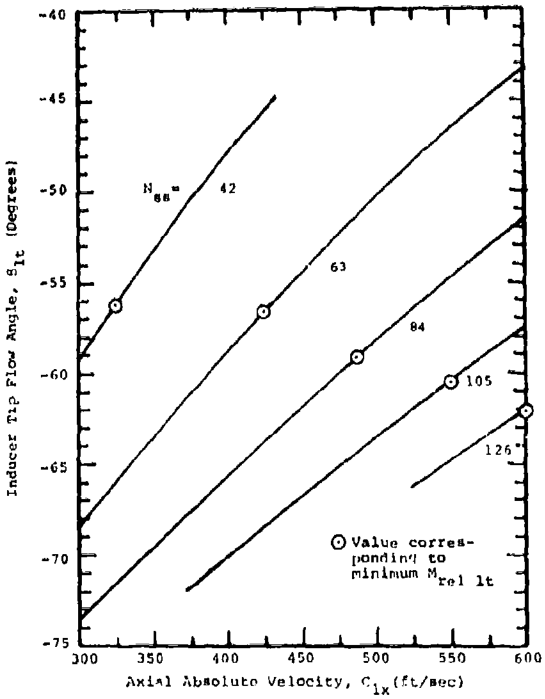

Figure 7. Inducer Tip Flow Angle Variations

We have not constrained the impeller exit/diffuser inlet velocity triangle to any fixed magnitude. Rather, we have set the exit flow angle (which refers to mixed out conditions) and have reduced the velocities to the lowest possible level by using maximum blade backsweep.

Generally, the diffuser exit velocity triangle is not of concern for stage optimization. In this particular case, the exit flow must enter a combustion chamber either by collecting the diffuser exit flow in a scroll and delivering it to the combustion can or by dumping the flow into a plenum region which feeds the combustion can. In either case, good combustion design requires well controlled inlet flow conditions, specifically, the absence of any swirl. Therefore, if the compressor discharge flows in to a plenum region, it may be necessary to use a deswirl cascade prior to final discharge. This final cascade can be configured to be a diffusing element or it can be followed by a simple annular diffuser to pick up a little additional recovery.

Similar considerations are in order for the radial inflow turbine. The first component in the turbine system is the nozzle. The nozzle can either be a scroll followed by a vaneless space (as is commonly used in the small diesel turbocharger design) or it can employ a nozzle cascade. The advantage to the former system is its comparative simplificty (low cost) whereas the latter system has the possibility for higher performance since Mach numbers are generally low accept at the cascade discharge. For the present design, we will consider the first configuration due to its simplicity.

There is an optimum size for this turbine inlet scroll. The velocities throughout the scroll are essentially keyed on the impeller tip speed since the rotor inlet swirl is set by the scroll. We prefer to design to zero incidence thus the angular momentum into the rotor is set by the rotor tip radius and tip speed. This angular momentum is imparted to the flow field at the scroll inlet. There is an optimum scroll inlet radius which represents a trade-off between the velocity levels within the scroll and the scroll dimensions. Thus the geometric scale of the scroll is one parameter which must be optimized for good turbine efficiency.

The turbine rotor inlet velocity triangle is set by the required work output (tip speed) and by the desired inlet relative Mach number. We desire to minimize the inlet relative Mach number in order to keep the relative Mach numbers low throughout the entire rotor. Here we are constrained by an inlet flow angle which we will restrict to 70°.

The turbine discharge/exhaust diffuser inlet velocity triangle is optimized basically by choosing a hub-tip ratio which permits good rotor design and good exhaust diffuser design. Having selected a zero exit swirl configuration and the rotor inlet relative Mach number, we have greatly restrained the impeller exit velocity triangle. We can manipulate the exit relative flow angle and passage height by modifying the hub-tip ratio. As long as the exducer tip radius is in reasonable proportion to the rotor inlet radius, we will not significantly impact rotor efficiency. On the other hand, a higher hub-tip ratio can facilitate better exhaust diffuser performance including more uniform inlet velocity distributions.

Finally, there usually is no restriction on the turbine exhaust diffuser exit velocity triangle, unless it is part of a regenerative or recouperative system.

Preliminary Design

The preceding paragraphs point out some of the basic design considerations which must be faced at the beginning of any preliminary design/optimization study. Reasonable values of the different parameters have been suggested above. From this base, near-optimum preliminary designs can be computed (with one-dimensional geometry definition computer codes). An example (for the centrifugal compressor) is given by Dean (1974). The optimum design of either the compressor or turbine stage cannot be practically effected by hand-calculation; a computer code is necessary to optimize several geometric subsections according to a variety of aerodynamic phenomena. When these are used to compute basic stage performance, we obtain the following preliminary results:

Table 4 Preliminary Overall Compressor/Turbine Design Parameters, 25 SHP SGT

Parameter	Compressor	Turbine
$\dot{m}$	--0.475 lbm/sec--	
N	----75,000 rpm---	
pr,er	3.87	3.56
inducer or exducer radii (r_h/r_t)	0.272in/ 0.907in	0.499in/ 1.248in
rotor tip radius	2.359in	2.700in
overall stage radius	4.30 in	5.00 in
rotor tip depth	0.112in	0.131in
stage efficiency, TS	78.8*	83.5*

*Good tolerances assumed.

Design Optimization

These levels of performance are good values to begin the optimization process. They lie above the performance goals set in Table 3 and presume good manufacturing tolerance control. They must be derated if a very low cost product is desired. This decision must be iterated through the entire system design/market evaluation. Yet it is possible to improve the performance further through intensive design effort and laboratory development. To begin this process, it is instructive to study in detail basic aerodynamic parameters pertaining to the preliminary compressor and turbine design; they are shown below in Table 5.

Table 5 Basic Gas Dynamic Parameters Controlling Compressor/Turbine Performance (25 SHP SGT)

Compressor Parameter	Compressor Value	Turbine Parameter	Turbine Value
Inducer		**Scroll**	
$M_{rel,1t}$	0.65	M_C	0.52
IKE	0.14	$\Delta\eta_{scroll}$	0.093
i_{1t}	3.0°		
Impeller (Exit)		**Impeller (Inlet)**	
MR2	1.42	$M_{rel,2t}$	0.40
ε_2	0.39	IKE	0.10
$\Delta\eta_{IW}$	0.024	$\Delta\eta_F$	0.009
$\Delta\eta_{DM}$	0.007	$\Delta\eta_{SF}$	0.004
$\Delta\eta_{BF}$	0.030	$\Delta\eta_{clearance}$	0.041
$\Delta\eta_F$	0.047	$\Delta\eta_{DM}$	0.001
$\Delta\eta_{WL}$	0		
$\Delta\eta_{clearance}$	0.010		
Diffuser		**Diffuser**	
DIKE	0.36	DIKE	0.08
B_4	0.08	B_3	0.05
C_{p2*-4}	0.078	C_{p3-4}	0.70
C_{p4-c}	0.67	$\Delta\eta_D$	0.017
M_4	0.85		
$\Delta\eta_D$	0.094		

It is further instructive to study Figure 8 which shows a breakdown of compressor loss parameters as a function of specific speed.

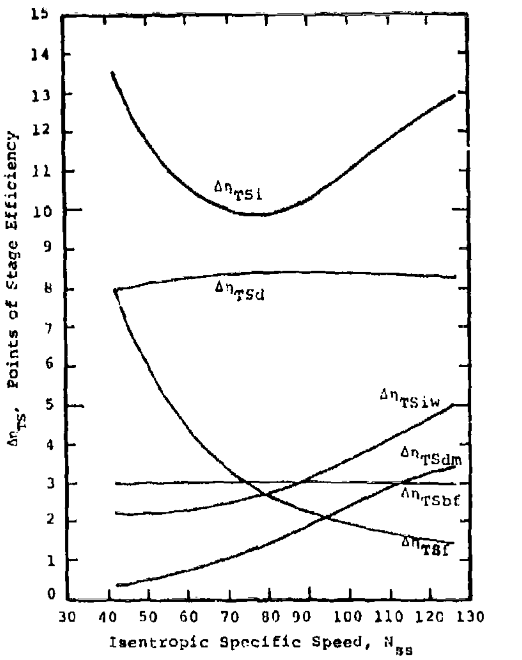

Figure 8. Variation in Fundamental Compressor Loss Parameters for a 50 shp small Gas Turbine Compressor.

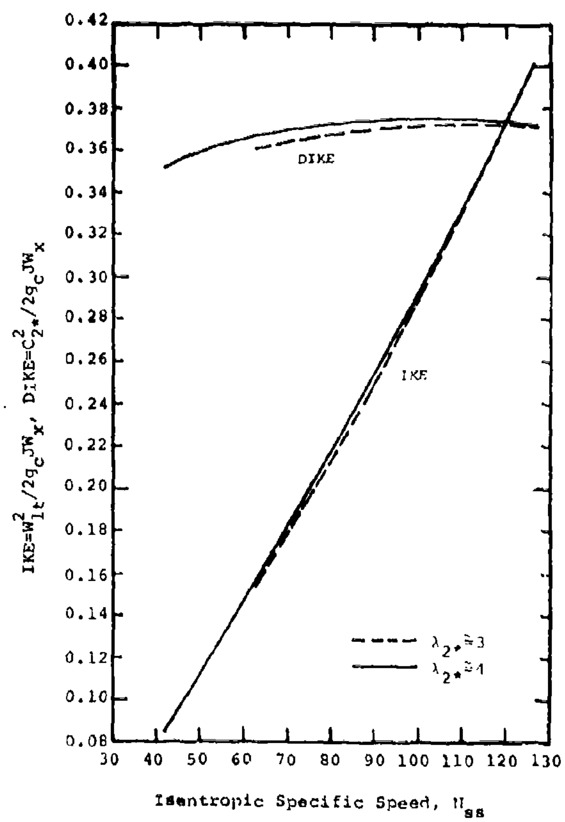

Figure 9. Distribution of Stage (Relative) Kinetic Energy in Terms of Work Input.

To improve the compressor, we can concentrate on a variety of the following areas:

1. Improved impeller performance including:
   a. increased relative diffusion (higher MR_2),
   b. reduced slip,
   c. reduced backflow loss (probably requiring changes in the vaneless and semi-vaneless space between the impeller tip and diffuser throat),
   d. reduced internal wake loss* (requiring preswirl in order to reduce $M_{rel\ 1t}$),
   e. reduced rear disk and shroud friction (requiring optimization of the rear disk cavity).
2. Improved vaneless and semi-vaneless space recovery including:
   a. reduced losses (we consider here only total pressure changes in the core flow which, in the absence of shock losses, implies large flow distortions caused by bad interactions between diffuser throat and impeller tip), and
   b. improved vaneless space diffusion (which implies either passage divergence or increased radius ratio).
3. Improvements to the channel diffuser recovery including the following:
   a. designing the diffuser passage closer to the ridge of peak recovery (on an area ratio versus L/W plot), and
   b. using sophisticated diffuser augmentation techniques (such as suction or passage shape modification for profile relaxation).

To optimize this preliminary compressor design, we must locate the most profitable areas for our attack. Figures 8 and 9 help us to select these areas. Invariably, we rule out preswirl as being mechanically impractical and therefore we can impact the internal wake loss only through MR_2 (or DR_2). It is difficult to significantly reduce rear disc and front cover friction penalties. We realize that the impeller and the diffuser are equally weighted with respect to opportunity for improvement (see Figure 8). It is also useful to understand where the recoverable kinetic energy in the system is. Therefore, we use the parameters IKE and DIKE which are the impeller relative inlet kinetic energy and the diffuser inlet kinetic energy respectively. These are defined and plotted on Figure 9. We see, at the design specific speed ($N_{ss,c}$=60), that the diffuser has a potent impact and that further improvements in impeller diffusion will be of limited value. It may be worthwhile to consider rear disc seals to cut friction losses. Probably the most practical way, under the restrictions assumed here, to improve the impeller is to reduce or eliminate backflow loss (which was set at 3 points).

Before turning to the turbine stage for efficiency improvements, it is perhaps appropriate to stress the role of IKE and DIKE. We use IKE to gain some appreciation for relative levels of kinetic energy in different pieces of turbomachinery. In Table 6 below are shown various levels of IKE and DIKE for a variety of systems.

* $\Delta\eta_{TSw}$ = $1/DR^2$ IKE $\dot{m}_w/\dot{m}$ for an incompressible fluid; represents energy lost when wake

Tab V	inetic Energy Distribution In Radial Flow Turbomachines			
System	Compressor IKE	Compressor DIKE	Turbine IKE	Turbine DIKE
ACV Lift Fan	0.27	0.40	-	-
Diesel Locomotive Turbocharger	0.20	0.43	-	0.21*
Truck Diesel Turbo	0.39	0.40	0.06	0.16
Small Gas Turbine	0.14	0.36	0.10	0.08

* Axial Turbine

We see that the values of IKE and DIKE change around significantly depending on the type of system under consideration. Therefore, our point of attack for stage optimization would be different depending on the actual unit being designed.

In turn, we can consider the turbine stage for performance optimization. We concentrate on the following areas:

1. Inlet volute design improvement (since the volute was optimized for minimum loss given the required velocity triangles, the only alternative is to switch to a cascade inlet),

2. Rotor improvement (the rotor has been designed to a nearly optimum configuration - we can improve it by changing the inlet incidence if a different distribution of efficiency is required at various off design points or by trying less acceleration in the actual blade passages, thus reducing the exit velocity), and

3. Exhaust diffuser recovery can be improved by modifying the basic diffuser geometric parameters such as inlet hub-tip ratio, $L/\Delta r$, divergence angle (including splitters) and so forth.

As with the compressor, we must carefully choose our areas for design optimization if we are to achieve maximum benefit. Again, understanding the relative levels of kinetic energy through the turbine is of use. The appropriate values of IKE and DIKE have been shown in Table 6 above.

Note that little direct attention has been given to losses; that is, loss correlations have not been dispersed through this study nor have we attempted to manipulate parameters which are constrained in a loss correlation. Rather, we have focused on the basic aerodynamic processes which either achieve maximum pressure recovery for a given element or efficiently convey fluid through an element without creating flow field distortions which engender high losses.

It is comparatively easy to reveal areas where performance improvement can be won; it is considerably more difficult to actually effect these improvements. Our previous discussions have used only one-dimensional concepts and computational tools to establish stage configurations. We can use these tools somewhat further in our optimization studies (particularly to find trade factors for different proposed design modifications) but will soon find ourselves using two-dimensional and three-dimensional computational tools and even the experimental laboratory to find design modifications leading to improved performance. A few examples are given below. The one-dimensional analysis approach is still of utility while seeking improvements in the compressor diffuser system. For example, we could consider a vaneless diffuser instead of a channel diffuser since the backflow loss would be eliminated. Vaneless diffuser performance can be readily computed with simple one-dimensional (multi-station) integration through the diffuser including appropriate wall shear stress levels for the particular piece of turbomachinery. Other diffuser types could be considered (conicals, double divergence, cascade, etc.). However, laboratory studies of conical and channel diffusers (see Runstadler *et al.* (1975)) and actual compressor test experience indicates that we cannot do better than an optimally designed straight channel diffuser (over conicals, cascade, etc.) A comparison between the vaneless diffuser and channel diffusers is shown in Table 7.

Table 7 Alternative Compressor Diffuser Configurations

Diffuser Type	$\Delta\eta_{bf}$*	C_{pD}	$\Delta\eta_D$*	$\eta_{TS,Stage}$*
Vaneless diffuser, $R_5 = 2.0$	0	0.410	19.5	70.1
Vaneless diffuser, $R_5 = 5.0$	0	0.510	16.3	73.9
Channel diffuser, AS_4=1.4, B_4=0.08	3	0.695	9.4	78.6
Channel diffuser, AS_4=1.0, B_4=0.06	1	0.705	9.3	80.7

We see that a vaneless diffuser is useless for this application. Furthermore, only small improvements can be made using a more efficient diffuser alone; the greatest gain to be had is by reducing the backflow loss which leads, ultimately, to an improvement in stage efficiency. Systematically reducing backflow loss is not a simple matter.

The actual flow process which exists between the impeller tip and the diffuser throat is a complex three-dimensional, time-dependent, viscous flow field. We can get a better grasp on the flow state by using two-dimensional (or quasi-three-dimensional) techniques which will compute the blade-to-blade velocity distributions. However, it is not at all clear that these calculations are of any real value since the flow field is very three-dimensional and unsteady. One can argue that the approach is useful for setting incidence; test experience fails to confirm this. Therefore, we are forced to move virtually from one-dimensional design stage directly to the experimental development laboratory where various design approaches, based on fundamental diffuser technology, can be experimentally examined in actual compressor test rigs.

We can similarly concentrate on other areas in the entire compressor and turbine system and follow a similar line of thinking to lead us toward optimized components. This must be done before any system can be considered to be truely optimized and at its best level of performance, given the current state-of-the-art and design restrictions (performance, cost, life). We will not pursue those further here since the approach is outlined above. Furthermore, we will not pursue the example design further since it has served its purpose: to illustrate the techniques used to achieve maximum performance.

The profile shape of the compressor and turbine, after the preliminary design and with some design optimization is shown in Figure 10. However, it is not representative of a final design which is ready for manufacture. Further optimization, perhaps even including some experimental development work, is in order to achieve truely optimum stage performance.

Efficiency levels of 78.6% and 82.5% (TS, adiabatic) are projected for the compressor and turbine respectively. Improvements of about 2 points each are possible with laboratory development. Good component tolerances are assumed.

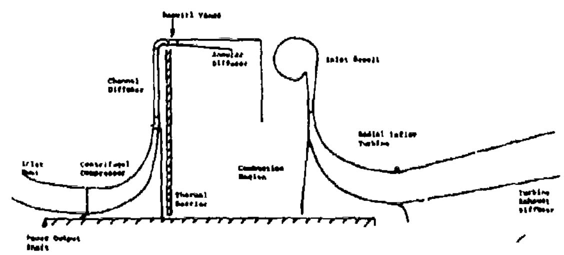

Figure 10. Preliminary Design Configuration for 11 HP Small Gas Turbine

5 PERFORMANCE MAP PREDICTION

Preparation of a complete performance map for all expected operating conditions completes the preliminary design effort. Actually, one cannot carry out the design optimization study and then turn to the present task with indifference. When the match point optimization is complete, the off design performance is completely set (unless variable geometry components are employed). Actually, without stressing the point, certain parameters were carefully chosen in the preliminary design and design optimization study to assure good operating range (particularly for the compressor). For example, the match point incidence was kept low in order to assure good flow range at reduced speed (50% - 70% speeds). In fact, if it was required to optimize SFC (or efficiency) along the entire gas turbine operating line, as may be required for an automotive gas turbine which would operate at a number of different speeds, then it would be necessary to choose several "match points" and consider optimization trades between the different so-called match points. Occasionally, we are required to go to this extreme. For the present study, we have followed a single match point optimization approach followed by off design performance prediction.

The prediction of off design performance is comparatively easy, contrasted to the match point studies outlined previously. Nonetheless, there are several difficult aspects. If it is necessary to predict a map with fidelty at very low speeds and low flows, we find ourselves reaching across a very wide spectrum of possible fluid dynamic states wherein basic flow phenomena can change radically. For example, the steepness of a compressor speed line, which is important for stage static stability, is impacted by many parameters including:

1. impeller relative diffusion characteristics,
2. slip factor variation with all relevant off design parameters (see Sturge & Cumpsty (1975)),
3. variation of backflow loss with all relevant off design parameters,
4. variation of Welliver loss, and
5. variation of diffuser static pressure recovery coefficients.

Unfortunately, the current state-of-the-art of centrifugal compressor design/analysis prohibits a proper treatment of all of the above parameters. For example, we have not been able to develop any rigorous modeling of the backflow fluid mechanics other than to assess nominal levels based purely on past experience. In fact, the subject matter goes virtually untouched in the centrifugal compressor literature. On the other hand, after years of

careful measurements, we have been able to accurately establish the relative impeller diffusion characteristics and slip factor variations, but the latter only in the absence of diffuser pressure fields at the impeller tip. Actually, it is not a bad assumption to take slip factor information from vaneless diffuser studies into the interaction case.

Thus we are faced with a very difficult situation in attempting to predict with sufficient accuracy the complete off design performance map. But the key probably lies in the term "with sufficient accuracy". In many cases, it is not important to be able to predict the entire map with complete precision. It is important to be sure that one can operate in the vicinity of the desired operating line (without choking or surging) and that reasonable efficiencies will result. It is also important to have tools that are sufficiently accurate that one can faithfully trust the models to lead in the direction of true design optimization. Based on success, we find we have been able to achieve these objectives very well with the models reported, for example by Dean (1974). Nonetheless, it is clear that of the above list of fundamental flow phenomena, not all are well preceived today. Therefore, one is forced to use simple models, for example for backflow, Welliver loss and slip factor when predicting a complete performance map. In fact, if we had an alternative it would surely lie in a complete three-dimensional, time dependent viscous flow field calculation which would be totally impractical for computing such maps.

Figure 11 shows the compressor performance map using a Weisner slip factor, a frozen level of backflow loss, zero Welliver loss and diffuser pressure recovery consistent with past measured experience and the data of Runstadler, et al. (1975). The surge line is labeled "conservative surge estimate" and represents a safe limit which is established by using the highest flow on each speed line where one of the following indications of stage instability is found. These indicators are:

1. zero speed line slope (i.e., $\partial pr/\partial m=0$),
2. impeller incidence has risen to the stall limit,
3. the level of vaneless space static pressure recovery has exceeded a critical value.

We know from experience that we can operate past any of the above conditions in certain cases without surging a stage. Nonetheless, it is impossible today to predict stage surge in a centrifugal compressor with precision. Therefore, for design purposes, we use the above conservative approach to establish the expected stable operating range. Generally, we can achieve greater range than that shown above.

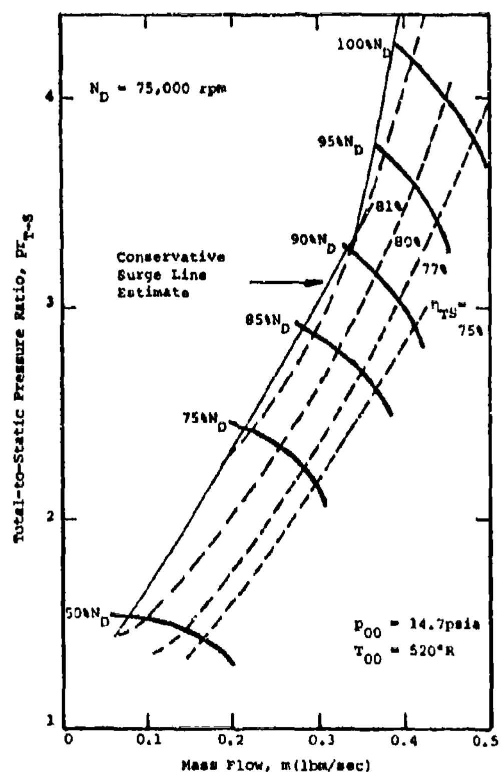

Figure 11. Preliminary Compressor Performance Map for a 25 SHP SGT. Reynolds Number and Clearance Penalties Included; Good Manufacturing Tolerances Assumed.

The off design turbine performance characteristic is shown in Figure 12. Here, there are fewer off design fluid dynamic phenomena which present performance prediction difficulties. However, one phenomena is worth stressing. When the turbine is operated sufficiently far off design, there are numerous opportunities for separated flow to occur in the rotor. For example, there is very strong tangential loading in the radial inflow portion due to the very rapid change in angular momentum with radius. In fact, the loading can be so great as to cause a separation bubble in this region even at the match point with zero incidence. Potential flow calculations even with zero incidence can show a potential recirculation zone. To make matters worse, as a turbine is operated further off design, it is possible to separate the exducer suction surface or hub surface. If off design performance is to be predicted accurately, it is necessary to represent these separated flows and to incorporate the important mixing loss which results. Unfortunately, a flow field which can have three or four different possible modes of separation is extremely difficult to predict, particularly on a one-dimensional station-by-station calculation procedure. It is doubtful if it is even worth going to more complicated calculation methods considering the cost and time involved.

An alternative approach to presenting the off design performance simply uses correlations of rotor performance to predict far off design characteristics. An example is the approach developed at NASA as detailed by Glassman (1976). This approach is quite adequate for developing a performance map for situations where measurements have been made on identical (or very similar) impellers and nozzles on previous occasions. Unfortunately, we

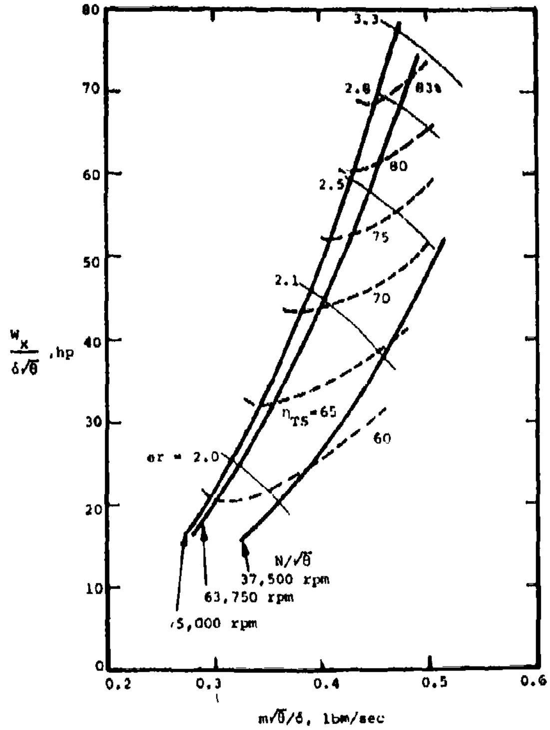

Figure 12. Preliminary Turbine Performance Map for 25 SHP SGT; Good Manufacturing Tolerances Assumed.

rarely find ourselves in opportunities where such information is available and useful. It is decidedly inappropriate if we are attempting to use our design tools to find new optimum systems.

The performance map shown in Figure 12 is therefore accurate in regions where the fluid dynamic states are reasonably well behaved. As one goes to extreme values of incidence (perhaps more than ±10° or 15°) we expect that the losses will develop much more rapidly than shown in that map. Incidentally, the separated flow problem also exists in the centrifugal compressor. We have been able to incorporate that phenomena in the performance map shown in Figure 11 without concern for off design performance prediction inaccuracies. In this case, the basic flow models assume that the flow is separated all the time (or that a significant portion of the flow field is of low momentum, i.e., heavily retarded boundary layers swept into a wake region). But the distinction between the compressor and the turbine appears to be in the various locations in the impeller where separation can occur. In our compressor modeling we treat a basic wake region which develops part way through the impeller and then passes through to the impeller discharge, followed by a complete mixing process. For the turbine, it is quite possible that the flow into the turbine could separate, be reattached and then leave the exducer with two or even three surfaces separated.

6 FINAL DESIGN OPTIMIZATION

The principal emphasis in this paper has been on the preliminary design and design optimization studies required to generate a high performance centrifugal compressor and radial inflow turbine. However, to complete the reader's understanding of the complete design process involved, it is appropriate that a few paragraphs be set aside to explain the subsequent detailed design steps.

The overall centrifugal compressor and radial turbine design approaches can be appreciated by studying Figures 13 and 14 which show the design flow chart. The reader will readily recognize many portions of the design process which have already been covered in the previous sections. The remaining steps in the design process involve two and quasi-three-dimensional flow field

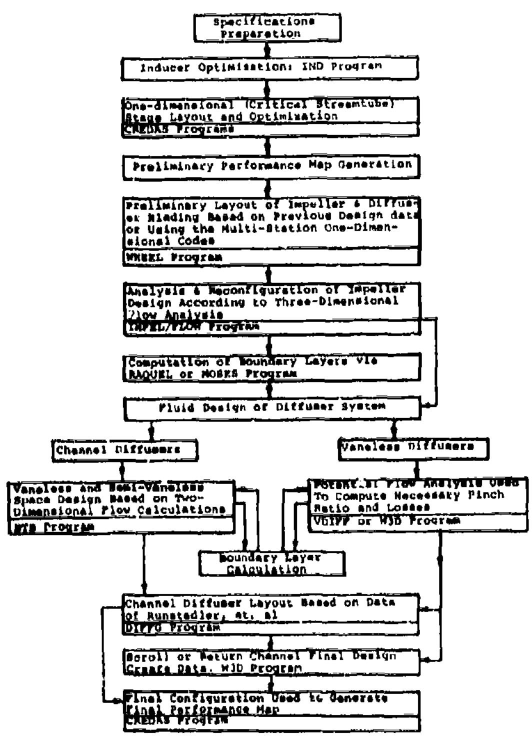

Figure 13. Creare Compressor and Fan Design Procedure

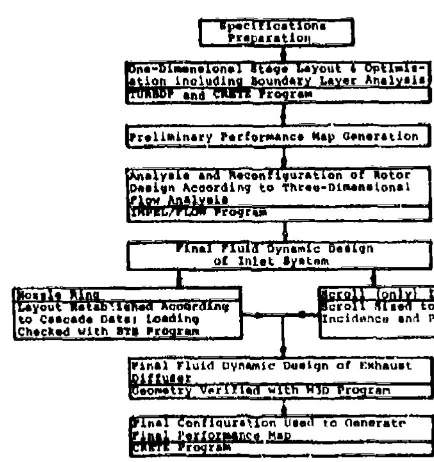

Figure 14. CREARE RADIAL INFLOW TURBINE DESIGN PROCEDURE

calculations. For example, quasi-three-dimensional flow field calculations are made for the impeller blade passages in order to develop suitable blade shapes yielding optimum loading characteristics. The approach we use to this work is detailed in a recent publication by Howard and Osborne (1976). A further example, of course, includes blade-to-blade two-dimensional flow field calculations as might be used in the turbine inlet nozzle (if employed) or in the vaneless or semi-vaneless space of a centrifugal compressor.

In the context of this presentation, we note that after the complete rotor blade definition work is finished, it may be necessary to recompute the complete performance map if critical design modifications have been made during the latter design steps.

7 CLOSURE

A highly refined approach to the preliminary design and design optimization of radial compressors and turbines has been presented. Particular emphasis has been placed on the use of fundamentally sound fluid dynamic models of critical turbomachinery processes. With this approach, it is possible to systematically optimize new designs for new applications with maximum confidence of developing new products which can make a significant impact in the market.

As an example of the techniques used, a paper study was presented of a 25 SHP SGT. While the particular machine has not been built, each element of the system is rather close to elements previously designed by Creare's staff for various industrial projects. Therefore, we have confidence that a system of this type with performance levels of the nature shown herein is practical. The sample case (25 SHP SGT) was based on a large number of system assumptions which must be modified from one application to another. System modeling and prediction considerations are the domain of a related lecture in the same series.

The author gratefully acknowledges the direct assistance of S. Wei and S. Everts; discussions with L. R. Young and C. Osborne have strengthened the work presented here.

NOMENCLATURE

AS_4	channel diffuser throat aspect ratio (b/w for radial-plane divergence; w/b for meridional-plane divergence)
B	boundary layer blockage: $B \equiv 1 - (A_{effective}/A_{geometrical})$
b	meridional depth of passage (normal to mean meridional velocity component)
C	absolute velocity (relative to a Newtonian frame, e.g. compressor casing)
C_p	pressure recovery coefficient: $C_p = (p - p_{ref}/(p_o - p)_{ref})$ where measuring and reference states and stations must be specifically defined.
c_p	specific heat at constant pressure
DR	diffusion ratio: $DR = (W_1/W_{2j})t$
er	turbine expansion ratio, total-to-static
g	proportionality constant in Newton's Second Law, $F = Ma/g$
i	incidence angle of flow onto blades: $(i = \beta_b - \beta)$
J	constant = 778 $ft\text{-}lb_f/Btu$
k	ratio of specific heats
L	diffuser centerline length (from throat to exit plane)
M	Mach number
m	mass flow rate
MR	Mach number ratio = $M_{rel\ 1t}/M_{rel\ 2j}$
N	shaft speed
N_{ss}	specific speed: $N_{ss} = N\sqrt{Q_o} / (\Delta h_{os})^{3/4}$ where N = rpm, Q_o = inlet flow = m/ρ_{oo} in $ft^3/sec.$, Δh_{os} = isentropic stage enthalpy rise in $ft\text{-}lb_f/lb_m$
P	shaft power
p	static pressure
p_o	stagnation pressure
pr	pressure ratio $pr \equiv p/p_{oo}$
R	radius ratio, r/r_2
R	gas constant
r	radius
T_o	stagnation temperature
u	impeller (metal) velocity
w	diffuser throat width (in radial plane)
W	relative velocity (in coordinate system rotating steadily in Newtonian space)
W_x	total shaft work per unit mass of fluid
X	bearing work factor, see equation 4
β	relative flow angle
ε	ratio of wake/passage widths; cascade turning
η	efficiency $\eta = \left.\frac{W_{xs}}{W_x}\right\|_c$ or $= \left.\frac{W_x}{W_{xs}}\right\|_t$
λ	swirl parameter: $\lambda \equiv C_\theta/C_r$
μ	work input coefficient: $\mu \equiv W_x/u_2^2/g_o$
ξ	vorticity or loss coefficient: e.g. $\xi_{IGV} = (p_{oo} - p_{o1})/(p_o - p)_1$
σ	slip factor: $\sigma = 1 - V_s/u_2$
χ	er/pr

SUBSCRIPTS

b	blade property or bearings
BF	back flow
c	compressor
coll	collector station (receiving volume after diffuser)
D	diffuser
dm	(rotor) discharge mixing
f	friction
h	hub
i	inlet or impeller
IW	internal wake (loss)
o	stagnation
rel	relative to impeller coordinates
s	suction surface
SF	secondary flow
t	tip or throat or turbine
TS	total-to-static (efficiency)
TT	total-to-total (efficiency)
WL	Welliver loss, ξ_{2*-4}
w	wake

SUPERSCRIPTS

*	mixed out state (must be specifically defined)

STANDARD UNITS Except as noted, all quantities herein have the following units:

C,W,u	fps
c_p	Btu/lb_m - °R
g_o	(lb_m/lb_f) ft/sec^2
m	lb_m/sec
N	rpm
p, p_o, p_t	psia
P	hp
q	psi
R	ft-lb_f/lb_m - °R
r,L,w,etc	inches
T, T_o, T_T	degrees Rankine (°R)
time	sec
W_x	Btu/lb_m
ρ	lb_m/ft
α, β	degrees

STATION NUMBERS

The Creare sign convention assigns all flow angles positive for velocity vectors that are in the direction of the wheel rotation.

STATION NUMBERS

Compressor

0	inlet (stagnation)
1	inducer inlet
2	impeller tip
3	vane leading edge
4	diffuser throat
5	diffuser exit

Turbine

0	scroll inlet (stagnation)
1	nozzle inlet
2	impeller tip
3	impeller exit
4	diffuser exit

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CHARACTERIZATION OF COMPONENTS PERFORMANCE AND OPTIMIZATION OF MATCHING IN JET-ENGINE DEVELOPMENT

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SUMMARY

Design and development of jet-engines require prediction and later, characterization through test analysis, of the performance of the engine and its components.

Knowledge of component characteristics is generally synthesized in mathematical models which contribute highly to efficient design and development.

At the beginning of development (i.e before first runs of prototype engines) models are essentially based on estimates and rig tests results. Problems are then encountered when engine tests results are compared to prediction.

Methods using engine test analysis to identify component operating characteristics as installed in the engine and leading to models more representative of aerothermodynamic behaviour of engine, are presented.

Such models appear to be very useful tools during the various phases of development. Application and coordination with tests are discussed and particularly relative matching of components and control schedules optimization.

RESUME

IDENTIFICATION DES PERFORMANCES DES COMPOSANTS ET OPTIMISATION DE L'ADAPTATION PENDANT LA MISE AU POINT DES TURBOREACTEURS

La conception et la mise au point des turboréacteurs nécessite la prédiction, puis l'identification par l'analyse des essais, des performances du moteur et de ses composants.

La connaissance des caractéristiques des composants est généralement rassemblée dans des modèles mathématiques dont l'utilisation accroît notablement l'efficacité de la conception et de la mise au point.

Au début de la mise au point (c'est-à-dire avant les premiers essais des moteurs prototypes), les modèles sont fondés essentiellement sur des estimations ou des essais partiels. Des divergences sont en général rencontrées lorsque l'on compare les résultats d'essais sur moteur avec les prédictions.

Les méthodes utilisant l'analyse des essais sur moteur complet pour identifier les caractéristiques de fonctionnement des composants tels qu'ils sont installés dans le turboréacteur, et conduisant à des modèles plus représentatifs du comportement aérothermodynamique du moteur, sont présentées.

De tels modèles apparaissent comme très utiles pendant les diverses phases de mise au point : les applications, en coordination avec les essais sont discutées et plus particulièrement l'adaptation relative des composants et l'optimisation des lois de régulation.

1. INTRODUCTION

From design to production of the operational engine, many studies and tests take place, or components, as well as for a number of prototype engines. Each phase of development requires more and more detailed knowledge of engine operation as it gives ability to appreciate qualities and deficiencies of the engine and components and leads to pertinent modifications to reach the specified performance.

That knowledge is derived from analysis of available experimental and theoretical data. Practical use of these data in a synthetic form requires creation of mathematical models. These models are recognized as the most valuable means of collecting information and an effective medium of communication since mathematics is the most common language of engineers. Use of these models during the design phase is well known as essential. Updated with test results, models can contribute highly to efficient development.

The Performance engineer's most usual configuration is an assembly of component submodels linked by mutual balance equations based on classical mechanical rules.

Modular approach provides many advantages :
- easy updating due to module independance
- clear understanding of engine operation as component contribution to overall balance is visualized
- common basis of component performance analysis with component experts.

Identification of in situ component performance as well as overall engine performance and stability can be derived from thorough analysis of engine test data. Development studies will be performed by model means after updating each component submodel with the results of test analysis. These studies lead to clear evaluation of modifications under consideration to cure deficiencies. In many cases simulation will give a good understanding of engine operation during test, particularly in flight.

In that process, the main difficulty is model updating and fitting with reality since the prediction model established for the design phase rarely appears correct when compared to test results.

The objective of this paper is mainly a description of techniques to relate engine test analysis to model updating.

Applications of engine and component status models will be presented.

2. PREDICTION COMINGS SHORT

2.1. Performance deviation and design margins

First engines installed in the bench are generally fully instrumented. Direct comparison of measured parameters to predicted values reveals immediately some differances.

It is extremely rare that a new engine reaches the specified performance at the first test and there is a simple reason for that : Design is always at the limits of present technology and ignorance factors are large.

In the aerothermodynamic field most frequent disappointments are :
. for compressors - surge line too low, lack of efficiency, and lack of flow at high rotational speed
. for turbines - efficiency and flow lower than target.
. burner often show inadequate temperature profiles.

Then it is logical that component designers attempt to introduce margins to limit such risks.

Sometimes these margins appear to be insufficient. But the opposite situation, when margins appear eventually to be excessive, may also be a handicap.

As a matter of fact component design is defined for optimal mutual matching after studies on the basis of component specialists data. If any one of the components has different characteristics than expected, whatever direction it shows, there is no reason for matching to remain optimal and close to specified conditions.

Thus , a rig tested component showing performance "in excess" of specification may be not so valuable if it induces mismatching of other components and then operation in a low performance part of their map. It is even possible that the faulty component puts itself in the situation where it is finally poorly used.

Margins cannot be substituted for ignorance factors. Advances in prediction techniques, analysis and reduction of these ignorance factors during development are the best ways to reach more severe and often conflicting targets.

2.2. Sources of prediction errors

Sources of error in prediction are :
. quality of theory and method (assuming data available)
. missing, wrong, or obsolete data.

Extremely complex methods are of no practical use if they require to much computer and data collection time to be carried on within manageable conditions. When costs and time are near each other, choice between paper prediction and test will be directed by effectiveness, accuracy and proof of the two approaches. At the present time rig tests are often preferred although they leave place to many potential differences with on-engine operation.

Let us quote :
1. Technological differences and mechanical environment.
On the rig a certain number of component features are not identical to the engine's, either because rig facilities requires special installation parts, or because the engine configuration has evolved.

2. Thermal environment differences.
Blade tip and seals clearances, and all temperature dependent mechanical deformations, are very important for modern high loaded turbomachinery. On-engine deformations are not reproduced exactly in the rig.

3. Reynolds number and temperature effects.
All rig test facilities do not allow inlet pressure and temperature change. So, viscosity losses variation are not determined. Moreover, change in characteristics may occur when physical speed or temperature varies at constant corrected speed due to item 2 effects.

4. Rig test facility features.
The rig test facility comprises special features as inlet duct and bellmouth, exhaust diffuser or volute struts, bearings. These features create particular operating conditions sometimes quite different from the engine's, as wakes, distorsion pattern, stress.

5. Components interference
Component isolated in the rig are not subjected, as on engine, to adjacent component interactions as temperature and pressure profiles, wakes, turbulence generated upstream and downstream.

Limitations of isolated component rig capability can be partially overcome by sophisticated test facilities such as :
. dual spool compressor rigs
. burner plus turbine hot gases rig
. core engine rig
. gas supply to these rigs with varied pressure, temperature, distortion and turbulence levels.

These very expensive facilities, often more difficult to operate than an engine, are of course very useful but do not eliminate on-engine component characterization.

Use of large portions of the engine in such tests leads very often to run it late after first engine tests. Except if there are crucial problems, development program management generally prefers to put the effort on engine tests.

In the following, component performance prediction will be considered as derived from rig testing analysis as well as from theoretical calculations. Let us see now what is involved in the problem of characterization of the engine and what are the experimental means available.

3. PERFORMANCE CHARACTERIZATION - ASPECTS OF THE PRACTICAL PROBLEM

3.1. Model and measurement compatibility

Each component having a predicted set of characteristics, to characterize that component performance in situ through engine test analysis means compare and establish differences with prediction in any point of the map of characteristics.

Fair and correct comparison requires careful definitions and assessment of parameters to be compared.

Modelling and analysis of the cycle is generally made assuming one dimensional streams. Basic characteristics maps of components are presented according to that assumption. Measurements at rig and engine bench show clearly that this assumption is somewhat artificial and that clear definitions of mean values are important.

Transformation of three dimensional measurement to one dimensional means values must satisfy the thermodynamic laws employed for the cycle balance calculations. Average pressure, temperature and velocity are usually derived from local measurement by application of energy, mass, and momentum equations to elemental streams. But some difficulties come up as these equations do not always preserve simultaneously flow, force and area.

Thus, use of one dimensional calculation has to be improved by fits like temperature profile factors, velocity coefficients, effective area coefficients. Conventions of that type have to be accepted by all parties concerned with the aerothermodynamics of the engine prior to measurement interpretation and discussion.

3.2. Observed components performance

First step of engine test analysis is determination of cycle, gaz flows and efficiencies of main components or group of components for each operating point described by simultaneous readings of measured parameters. Then straight comparison to predicted values can often give a misleading picture of component quality. As said before mismatchir· ıe to characteristics different than predicted, causes irregular operation of each component.

That "migration" as it is commonl) . ., superimposes its effect to proper anomalies of the component and it must be clearly discriminate.

Within the corrected one dimensional scheme adopted and migration effect being separated, the difference between prediction and engine test analysis results can be attributed to the factors listed in 2-2.

Contribution of each one in the total difference will have to be quantified in concurrence with component designers. This difficult operation may require further test either on the engine or the rig. Although it is a master key for progress in prediction ability, it is not always considered as an essential part of the development program. Often the performance engineer will have to accept many of the differences derived from his analysis without full explanation.

3.3. Experimental availability

3.3.1. Test facilities

The performance engineer's wish is to identify a component in the entire part of the map used by engine operation.

Generally engine test cannot cover easily the whole map of each component. However, engines equipped with variable or adjustable features such as nozzle exhaust area, turbine nozzle, air bleed, power off-takes, bleed back in flow path, compressor stator vanes and adjustable fuel flow control system are more frequent with modern technology and these configuration offer quite acceptable capability to explore components maps.

Simulated altitude facilities or flying test bed give the opportunity to get tests results with inlet conditions covering the flight envelope. Thus Reynolds number and thermal environment variations, high corrected speed, high mechanical speed, high expansion ratio in the nozzles can be studied.

Even if variable features are not normally included in the production definition, it is worthwile to use it during the development program at least on some specially assigned engines. Wise like use of inlet distorsion generator and fuel flow-former will contribute strongly to transient performance and stability evaluation.

3.3.2. Instrumentation

Instrumentation quantity and position depend on technical possibilities of installation and environmental conditions as temperature and vibration levels.

Thus it is easier to install rakes of pressures and temperatures probes upstream and downstream of compressors than in turbine nozzles or in after-burner exhaust nozzles.

Number of rakes in a reference station is a compromise between minimum number of probes to obtain a good representation of flow conditions, maximum number of bosses compatible with mechanical integrity of casings and limited perturbation into the stream. Rake angular positions are usually defined to cover pitch of the last upstream vanes. Radial position of probes is usually set to cover constant area or constant flow rings. Such a distribution gives generally an accurate approximation of average values of pressure and temperature by arithmetical mean.

Static pressure measurements are frequently restricted to wall taps except when local curvature of the stream is large and when flow integration is desired with great accuracy (i.e. total air flow in bell mouth).

Other classical thermodynamic measurements are rotational speed, flow and temperature of fuel, variable nozzles areas, variable stator angle, bleed valve position, thrust. External conditions as inlet temperature and pressure, flight or simulated flight speed, and ambient static pressure have to be also accurately measured.

Cavity pressures and temperatures in the vicinity of flowpath and metal temperatures help to determine parasitic and cooling flows.

Dimensions of physical areas in the flow path, seals and blade tips clearances and generally mechanical and geometrical definition of the engine tested have to be well documented.

Figure 1 shows typical instrumented stations of a turbofan.

Provided with these data, the performance engineer should be able to deal with cycle analysis; but first of all, elimination of erroneous data will give him some difficulties. As deep analysis requires a great number of measurements there is an inevitable part of wrong data. Very often inconsistancy of cycle balance will give the best evidence of measurement error. After elimination of erroneous data, reduction will give station vectors of average values for parameters to be input in quasi one-dimensional cycle analysis.

4. CYCLE TEST ANALYSIS AND COMPONENT PERFORMANCE ISOLATION

4.1. Cycle derivation for each set of readings representing test operating point must be worked out with calculation methods.

Obtention of a consistent cycle for each test operating point, raises variable difficulties depending on cycle structure and complexity, and number of instrumented stations. It is not always possible to install measurements in every important station marking component boundaries. So stream conditions there, have to be derived from balance equations. Inversely, to solve these equations, some data redundancy may occur, leading to several ways to calculate some of the characteristic parameters of cycle and components. Idealy these various solutions should be identical. Usually they are close but perfect consistency is never achieved particularly in early analysis.

Inconsistency is due to :

a) measurement accuracy
b) approximate representation of the theoretical scheme of calculation i.e model definition
c) estimate error in predicted data when not directly measured.

These estimated data are usually derived from theoretical sources, book-keeping of previous experience, rig test, or analysis of special measurements on the engine.

Let us quote :
. cooling flows, controlled or not leakage flows
. flow path physical or effective areas

- accessories and parasitic power and bleed off-takes
- some aerothermodynamic characteristics of some components assumed rightly or wrongly to be well established after rig test or theoretical prediction and reputed as little sensitive to other components interactions.
  For example : combustion efficiency map, pressure loss factors for burner and ducts.

After deep check of measurements it is also necessary to review all assumptions of that kind and adjust it, if possible, until the different methods of cycle balance agree within acceptable accuracy. The example to follow will permit an easier description of the various methods of cycle test analysis.

Figure 1 shows flowpath instrumentation capability of a typical modern engine.

Elements exhibited on figure 1, plus estimated data, lead to several sets of equations to establish a balanced cycle and derive components efficiencies.

As primary flow determination is generally a key-point in by pass engine test analysis, each method is distinguished by the method of calculation of that important parameter.

Figure 4 characterizes four typical methods :
1. High pressure turbine flow function
2. Energy balance
3. Low pressure turbine flow function
4. Fan duct flow function

For separated flows engine without reheat, nozzles flow function may also be a useful line of calculation.

Methods 1 and 2 are the most current. 3 requires inter turbine pressure measurement not always available as they may be difficult to install. Even installed, use if often hazardous due to very distorted profiles in that station.

Method 4 is of some interest for low by pass ratio engines but requires a well calibrated fan duct.

Various cross-check methods are applied through continuity equation when static and total pressure are available in stations of well known effective area.

Use of the whole set of available methods and cross check is worthwile mainly when first test analysis is concerned. Later, when consistency of the various calculation lines are well established by eventually corrected assumptions, use of a single method is possible. Then it has two advantages : shortened calculation time and unique referrence to compare small variations of performance.

However when accurate characterization in absolute terms is required, simultaneous use of the various methods and matching of assumptions to get their concurrence is an essential part of performance test analysis. Difficulty lies in the fact that inconsistency is generally sum of many small effects. Fortunately they are small enough to justify application of linear calculation techniques to determine which corrections will compensate.

A current process is determination of measurement or estimated data corrections by variations calculation means.

Thus derivative tables giving linear influence of each variable parameter on the results of the various methods is a very useful tool. However pertinent use of these tables in a trial and error process is a skill-proving technique.

As a matter of fact all that tedious work is nothing but a search for a single, although approximate, solution to cycle balance equations. Ideal solution could be considered as the one giving a vector of results at the minimal distance of each solution of the various methods just described. The computerized process leading to such a solution begins to be devdopped on the basis of regression method to minimize inconsistency. More details on that type of approach will be given next in connection with direct identification of engine model by "optimal simulation method".

Assuming now that a cycle balance is well established within acceptable accuracy, performance of each component comes out for every analysed test point in terms of flow, pressure ratio, temperature rise, efficiency.

riable configuration parameters such as variable nozzles allow to attainment of such results for different operating lines covering a portion of each component map. The second phase of analysis will be comparison to predicted or specified maps.

4.2. Component performance isolation

A comprehensive computer program for test analysis is able to produce all internal engine performance parameters and particularly those which are generally used for component performance characterization.

Each component map is defined by values of dependent parameters as functions of some chosen independant parameters. For example corrected rotational speed pressure ratio and inlet Reynolds number index may be the independent parameters for a compressor, and mass flow, efficiency, temperature rise dependent ones.

From the analysis of an engine test point is derived a set of values for dependent and independent characteristic parameters of each component. Comparison of that set to the prediction for the same values of independent variables is the only valid method as it isolates the component, and eliminates migration effects aberration.

When variable geometry on the engine or test facilities has permitted the exploration of a large part of the component map, it is possible to compare on-engine map to the predicted one.

Figure 3 shows a Fan map of a turbofan as derived from engine test at S.L.S. in comparison to a predicted one from rig tests. On-engine mapping was obtained with a variable exhaust nozzle.

It is not possible to cover a turbine map so largely. Figure 4 shows a predicted map and the portion explored by variable exhaust nozzle means.

Variable stators on compressor may allow a wider variation in corrected speed at constant load.

Likewise, some components have performance characteristics very dependent on pressure conditions and cannot be evaluated without altitude facilities. Figure 6 shows predicted variation of reheat efficiency with altitude conditions (when installed in the "predicted" engine).

Thus in some case, characterization of component performance by engine test analysis does not permit determination of the whole map. Updating component models will require extrapolation of the characterized portion. Prediction will then remain the best means eventually after correlation of divergences.

4.3. Variation method and correction factors

4.3.1. Use of variations is a good way to emphasize, analyse and correlate divergence from prediction. To quantify clearly that divergence, it is practical to define correction factors which will scale the prediction to fit test results. Moreover it makes a direct connection with the engine math model where scaler (or adder) availability is generally provided to work out design studies with modified component maps.

Correction factors derived from test analysis appear as measurement of a prediction ignorance factor. For each component a correction vector to scale dependent variables has to be carefully defined : correlation potential i.e reduction of ignorance capability often depends on a pertinent definition.

Let us review some usual component map transformations.

4.3.2. Fan and compressors

The thermodynamic process in a compressor can be represented by four of the following parameters :

- Corrected rotational speed
- Corrected mass flow
- Efficiency
- Enthalpy rise
- Pressure ratio

Two of which beeing considered as independent.
Then it is logical to define a correction vector with four factors related by 2 equations.

A typical compressor correction vector is :

- G1 = WR/WR reference
- G2 = Eis/Eis reference
- G3 = NR/NR reference
- G4 = ΔHR/ΔHR reference

alternates for G4 are often PR/PR reference or (PR − 1)/(PR reference − 1).

Choosing two of the G factors as independent, it is possible to establish relationships between the four factors to define various transformations of the map. Based on the fact that the G factors are close to 1 these relationships may be simplified.

For example modification of mass flow and efficiency at constant speed and pressure ratio will be defined by G1, G2, G3, G4 with G2.G4 = 1, G3 = 1.

Table 1 of figure 5 shows various transformations.
Choice of a type of transformation must be directed by the suspected main physical cause of modification. For example if divergence in flow and efficiency is not related to a change in the velocity triangle, G4 may be kept equal to 1.

4.3.3. Turbine

Most usual coordinates (i.e independent variables) for turbine maps are ΔH/T, WR.NR
or ΔH/T, NR

Dependent variables are pressure ratio, efficiency, exit angle, power etc...
As modern turbines have many cooling and parasitic flows, referred mass flow, specific load, and efficiency become very conventional. Generally mass flow is referred at the first nozzle vanes throat or a inlet of first blade row.

It may be practical to use power as the independent parameter and represent the turbine map in the following way :

- PR VS NR, PWR/PWR max
- WR VS NR, PWE/PWR max
- Eis = f (PWR/WR, PR)

and define correction factors as :

- G1 = WR/WR reference
- G2 = Eis/Eis reference
- G3 = NR/NR reference
- G4 = PW/PW max/(PW/PW max) reference

Like the compressor some relationships between these factors allow the definition of various transformations of the map. Table 2 of figure 5 lists some of them.

4.3.4. Other components

Ducts − Duct pressure loss is generally characterized by a single coefficient known as pressure low factor.

Pressure loss is then assumed to be proportional to dynamic pressure in a reference station in the duct.

In some cases ducts are intricated enough to have an overall pressure loss which does not follow so simple a law. There, a better model would be the sum of consecutive elementary losses each one having its own loss factor. Also, the engine duct loss may be affected by profile and wakes generated by the component supplying the flow. For that reason, pressure losses measured on the engine may lead to a variable loss factor compared to prediction. The correction factor may be defined as G = Loss factor/Loss factor reference.

Burner - Aerodynamic pressure loss can be treated as in a duct. Combustion efficiency is difficult to derive accurately from engine test. So in most cases it is assumed identical to rig test values.

Reheat - Thrust and exhaust nozzle area measurements lead to derivation of reheat system pressure loss and reheat combustion efficiency. Pressure loss and mainly efficiency may be complex functions of system configuration, reheat pipe inlet conditions, fuel air ratio and fuel distribution.

Predicted performance generally account for these parameters and it is possible to define correction factors in reference to that predicted map.

4.4. Interest of correction factors

Thus it is always possible to define, for each component, a vector of correction factors. Each analyzed test point leads to corresponding values of G factors.

Interest of these parameters appears when a whole set of values for a given factor gather in simple correlations of relatively small scatter and amplitude. This case is quite frequent as prediction quality has improved largely with modern methods. When these conditions are encountered it is possible to derive simple functions to represent G factors variations if there are any significant ones.

Input of these functions in the math-model of the engine will update it without requiring the refitting of the whole set of maps in the computer program. This can be considerably time saving.

Moreover if the explored region of the map is not large enough to update the whole map, necessary extrapolation will be easier on the basis of G factor correlation.

Generally, correlation has to be carefully established; statistical methods may be of good help when a large number of test points is available.

Figure 7 show correlation derived for mass-flow and efficiency scalers of the Fan map presented on figure 3, after analysis of sea level static test. G factors functions have been correlated to corrected speed which implies that divergence from prediction is due to internal physical differences with prediction configuration (in that case rig test); an assumption which appeared quite justified.

For the same component, simulated altitude engine test gave another correlation for extra mass flow correction versus inlet total temperature. Predicted Reynolds number effects were recognized as correct within measurement accuracy. Correlation with inlet temperature was explained by thermal expansion and for a larger part by untwist of blades when physical speed increases with T2 at fixed $N/\sqrt{T2}$.

In that example, upstream and downstream instrumentation gave good isolation of the component and correction factor identification was rather easy. This is not the case for all modules of the engine. Then results are more sensitive to measurements and assumptions errors.

Figure 8 shows scale factors derived for the efficiency of the turbine map presented in figure 4. Scatter is consistent with accuracy of the measurements involved in derivation of efficiency and other turbine parameters.

Figure 9 shows for the same turbine, correlations on efficiency obtained for various modifications tested on the engine. These results were all obtained by means of a variable exhaust nozzle and fuel flow modulation; so they are covering most of the operational region of the map.

Expected trends were qualitatively verified, however quantitative effects were somewhat different and confirmed that such modifications had to be evaluated on the engine.

These tests showed that large divergence from prediction at part speed was not due to environment effects but to prediction method shortcoming. Recent and more sophisticated methodes confirmed that analysis.

When amplitude of corrections is large and correlation is difficult to establish clearly, use of G factors is of little interest. When that situation occurs, it means either that prediction is too bad to be kept as a basis or that the real component shows a serious design or construction problem. Then it is rather recommended to reestablish new maps eventually after rig test evaluation.

Variations of 5 % must be recognized as limits for valuable use of corrections factors.

When correction vectors are determined for each component, they can be fitted in the engine model. Such an updated model has the advantage of integrating information derived from prediction, rig tests and engine test. Of course this method does not completely relieve the refitting of some component maps when new informations or new definition of the component justify it. However, time always comes in the development when components are no more extensively modified to require new rig or new prediction. Minor but often decisive modifications to overcome the last difficulties before reading target have to be evaluated on the engine.

Correction factors methods of updating is also applicable to establish a model of an average engine of a group of the same definition or a specific model of a particular engine of that group; then variations of the G factors have the magnitude of production tolerances.

5. DIRECT DETERMINATION OF CORRECTION FACTORS : OPTIMAL SIMULATION

5.1. Correction factors are especially suitable for cycle test analysis with a programmed system based on a prediction model.

As a matter of fact direct identification of the corrections and updating of the basic model can be worked out by automatisation of the classical process

Simulation - comparison - correction

where simulation of test results and operation of the engine is conducted with the original model.

Then comparison with real test results determine correction factors which are fitted in the model to update it.

Application to each test point will lead to a new model locally fitting the tests results.

Derivation of G factors require an iterative process until the corrected model reproduces all the measurements with the best possible accuracy. It appears that the regression method is easily adaptable to the usual model program.

Measurements and balance equations are often in a greater number than independent variables (i.e G factors) as it has been discussed in 4.1. When it is actually the case, equations to minimize simulation errors have to be substituted to exact classical ones. In most modern math models, the non-linear equations are solved by the Newton - Raphson method. Structure of the model is designed to allow a variable choice of independent parameters and equations in equal but variable number.

Non linear set of equations to be solved is :

$$\left.\begin{aligned} y1\ (x1,\ x2, &\ \ ,\ xi,\ xp) = 0 \\ y2\ (x1,\ x2, &\ \ ,\ xi,\ xp) = 0 \\ yj\ (x1,\ x2, &\ \ ,\ xi,\ xp) = 0 \\ yp\ (x1,\ x2, &\ \ ,\ xi,\ xp) = 0 \end{aligned}\right\}\quad \vec{Y}(X) = 0 \qquad (1)$$

Solution is calculated by an iterative process.
Each step gives an approximate solution that is derived by linearization of equation (1).
That approximate solution is $\vec{X} + \delta\vec{X}$ where

$$\left\|\frac{\partial y}{\partial x}\right\| \delta\vec{X} = -\vec{Y}(X)$$

Matrix $D = \left\|\frac{\partial Y}{\partial x}\right\|$ of partial derivatives is calculated by differences εX on each X and resulting one on each y.
Calculation is repeated until $\vec{Y} = 0$ is satisfied with acceptable accuracy.
Assuming now that each component submodel has the capability for G factors it is possible to choose some of them as $\vec{X}$. On the other hand simulated values $\vec{P}$ of measured parameters $\vec{M}$ lead to a set of equations (2) $\vec{Y} = \vec{P} - \vec{M} = 0$, with $\vec{P} = f(\vec{G})$ by model calculations. Approximate solution to $\vec{P} - \vec{M} = 0$ with minimum error is defined by the equation of minimum distance from $\vec{P}$ to $\vec{M}$.

$$L^2 = (P1 - M1)^2 + (P2 - M2)^2 + \ldots\ldots (P_p - M_p)^2 = \sum_1^p (Pk - Mk)^2$$

Set of equations to be solved becomes :

$$\left.\begin{aligned} \frac{\partial L^2}{\partial G_1} &= 0 \\ \frac{\partial L^2}{\partial G_2} &= 0 \\ \frac{\partial L^2}{\partial G_n} &= 0 \end{aligned}\right\}\qquad (3)$$

with n unknown quantities G.

It is important to consider as included in P (or M) not only directly measured parameters but also estimated parameters as it was discussed in 4.1.

Equation (3) can be modified in order to account for the relative level of confidence which can be attributed to each one of the measured or estimated values M. For measured parameters this level can be considered as proportional to accuracy of measurements. A weighting factor Ak is then associated to each term of the distance L, transforming (3) in :

$$\left(\frac{\partial \sum_1^p Ak\ (Pk - Mk)^2}{G_i}\right)_{i = 1,\ \ ,\ n} = 0 \qquad (4)$$

Solution of equation $\vec{P} - \vec{M} = 0$ beeing approximate, it is logical to exclude of (2) equations which must be solved exactly as mass flow, energy, and momentum conservation. If it is not possible by a local iterative loop it will be acceptable to treat it as equation (2) provided that the corresponding weighting factor be relatively high.

The Newton-Raphson method requires particular logic in the model program to calculate the partial derivative matrix :

$$\frac{\partial Yj}{\partial x_i}$$

This logic will be used identically to calculate $D = \left\|\frac{\partial\ (Pk - Mk)}{\partial Gi}\right\|$ but D has p lines and n colums instead of beeing square us usual.

If $\vec{G}o$ is a set guessed values a new approximate set $\vec{G}o + \vec{\delta G}$ to solve (4) is obtained by the linear regression formula :

$$\vec{\delta G} = |D^T . A . D|^{-1} \; D^T . A . (\vec{M} - \vec{P}) \qquad (5) \quad \text{where}$$

superscript T means "transposed" and A is the diagonal matrix of the weighting factors.

This calculation will be repeated up to the point where Σ Ak $(Pk - Mk)^2$ variation remains lower than a fixed value.

Stability and validity of results

The method presented here has the great advantage of integrating test analysis and improvement of engine and components models validity. However from the theoretical point of view it may seem risky as it requires a well marked minimum to converge. Physical nature of the problem and pertinent choice of measured parameters lead fortunately to stable solution in most cases. Anyway choice of convergence criteria is not clear.

Experience shows also that residual errors for Pk - Mk are often lower than $\sigma^2 k$ (σ k beeing the variance of measurement Mk) if weighting factor Ak was choosen proportional to $1/\sigma k^2$. That result is encouraging for development of the method. In any case if residual errors were large it will direct investigations toward measurement error, assumptions error or modelling shortcoming.

5.2. Correlation of correction factors - Possible extension of optimal simulation

Analysis of a number of test points covering a large part of the operational range leadsto a corresponding number of values for each G factor. As discussed in 4.3 it is necessary to establish pertinent correlations between that correction factor and some usual characteristic parameter of the concerned component or adjacent components. With modern computer means, it is possible to process an exhaustive search for the best correlation among many candidates. Regression methods may also be used successfully, to look for function reproducing the variation of G factors with minimum error.

If physical causes of corrections are well known it may be possible to define the functional form of the correlation prior to test analysis. Then unknown parameters become the constant coefficients of the function. Direct determination of that constant coefficients simultaneously for the various G functions is theoretically possible by utilization of the whole set of test points and a simulation-identification process derived from Kalman filter type method. However it is clear that memory size and computation time required is hardly manageable.

6. APPLICATION OF STATUS ENGINE MODEL

6.1. Use of the methods just described leads to an updated math model of the engine and its components with the ability to perform simulation of the status performance with an accuracy often competitive with real experiments. This ability joined to the extremely low cost and delay of computation compared to real test justifies the increasing interest of that tool.

Application can be divided in two groups : we shall mention both and discuss the second one. In the first one are applications similar to those completed with early prediction model. A status model will contribute to check or update design optimization, direct and prepare tests by preliminary simulation.

- Prediction of in flight operation for the first flight engines is of major interest. Post test or on line simulation help to check measurements and to understand engine behaviour. All these applications are based on a status engine and systems which may be still far from specifications.
  The second group contains applications contributing to improvement of engine and components in order to achieve specified performance As this requires that every element be pushed to its maximum efficiency, the first application will be studies of :

. Minor modifications of some components with low predicted risk of self degradation but advantage of rematching other components to optimum operation.
. Better control system schedules to make the best possible use of the engine.

Further contributions to development will be evaluation of the effects on the whole engine of substantial modification of one or more components. These modifications can be dictated by mechanical as well as by thermodynamic reasons. Of course variation of characteristic of the component to be possibly modified will be predicted.

Simulation of the modification can be incorporated in the model while keeping consistancy with the unchanged and well known parts of it.

Results of these studies will direct decision for the best choice in development orientation.

Of course the best solution to reach all the specified targets would certainly be to bring each component up to fit the characteristics on which specifications were based. However it is difficult to succeed for evident reasons as :

- Some objectives are proving to be unaccessible due to unrealistic prediction for specifications
- Valuable modifications for some objective are often conflicting with others
- If some solutions exist time schedule and cost eliminate most of it.

So studies aiming to determine means of improvment, while most of qualified components or part of component remain usable, are of prime interest. It must be remembered in any case that there is no substitute for quality; then it is unrealistic to pretend that objectives are attainable if none of the components has capability of compensating deficiencies of others.

6.2. Engine rematching

6.2.1. Practically, rematching means to introduce in the engine generally minor modifications of some components in order to set the operating line in the best place for performance and stability optimization.

These modifications are particularly valuable when they do not change the quality i.e efficiency of the component.
If any loss occurs it has to be compensated by better operation for another component.
Objectives of rematching are usually :
- Reduction of thermal load to reach specified thrust
- Reduction of thrust specific fuel consumption
- Rematching of airflow compatibility with air intake system
- Adequate surge margin to insure transient performance and distorsion tolerance.

The following table lists the various part of components on which actions are usually undertaken and their effects on the matching of a twinspool turbofan.

TYPICAL TURBOFAN REMATCHING ACTIONS

Component	Modification	Main resulting effect on component characteristics	Resulting effect on engine matching at constant turbine entry temperature	Resulting effect on engine performance
HP Turbine	Nozzle vanes stagger (open)	Increase flow capacity.	Decrease HP rpm, HPC airflow HPC pressure ratio, HPT load Increase HPC surge margin.	Improve starting and accel capability. Increase SFC at thrust.
LP Turbine	Nozzle vanes stagger (open)	Increase flow capacity.	Increase HPT load, HP rpm and HPC airflow. Increase HPC and LPC surge margin. Decrease LP rpm and LPT load.	Improve HPC and LPC surge margin. Increase thrust at TET.
Nozzle	Exhaust area (open) Nozzle angle (decreased)	Increase in effective area.	Increase LP rpm, LPT load Increase Fan surge margin and Fan flow. Decrease LPC surge margin. Increase reheat pipe Mn.	Increase LP rpm and thrust at TET. May change SFC plus or minus depending on efficiencies migration.
Fan	1st stage blades stagger or twist, or inlet guide vane stagger (open)	Increase flow at high corrected rotational speed.	Increase by pass ratio at high corrected speed. Decrease LP rpm.	Increase reheat thrust at altitude low Mn and fixed Fan surge margin.
	Last stage stagger or twist (closed)	Decrease flow at low corrected rotational speed.	Decrease by pass ratio at low corrected LP rpm.	Increase reheat thrust at altitude high Mn, fixed TET and Nozzle size.
			Increase LP rpm.	Increase LP rpm. May improve SFC at part power.
LP compressor	1st stages blades stagger (closed)	Decrease primary flow at high corrected rotational speed.	Increase by pass ratio at high LP corrected rpm.	Increase TET at altitude, low Mn, fixed Fan surge margin.
	Last stage blades stagger (open)	Increase primary flow at low corrected rotational speed.	Decrease by pass ratio at low LP corrected rpm.	Increase reheat thrust at altitude high Mn, fixed TET, Nozzle and LP rpm. May improve SFC at part power.
HP compressor	1st stages vanes and blades stagger (open)	Increase flow at high corrected speed. Lower surge line at low corrected speed.	Decrease HP rpm at high corrected TET. May improve surge margin at high corrected rpm. Decrease surge margin at low corrected rpm.	Decrease HP rpm and mechanical load. Performance depending on efficiency and surge margin change.
	Last stages blades stagger (open)	Increase flow at low corrected speed. Rise surge line at high corrected speed	Decrease HP rpm at lower corrected TET or corrected LP rpm.	

6.2.2. Two topics are often directing rematching actions on compressors and turbines of military turbofans:

By pass ratio adjustement

Higher Fan pressure ratio and airflow may increase thrust but a too large by pass ratio is conflicting with high pressure in Fan and exhaust pipe. At high flight Mach number, for a fixed reheat temperature and maximum nozzle area, the higher the pressure is, the higher the flow and thrust will be. With limited turbine entry temperature this requires low by-pass ratio. On the contrary surge problems at high corrected rotational speeds (i.e high altitude low flight Mach number) requires moderate Fan pressure ratio which is compatible with larger by pass ratio. As natural evolution of by pass ratio is unlikely following that trend, it may be worthwhile to match front and back stages of compressor and spools speeds to have the best by pass ratio variation.

Part power SFC improvement

Generally military engines have stringent specifications for part speed SFC. But to obtain good performance at low thrust as well as at maximum rating, is often very difficult. Variations of compressor and turbine efficiency with rotational speed are very important. At low speed, compressors have generally better efficiency than at high speed. It is the contrary for turbines. Moreover at low speed, compressor efficiency gradient is smaller than the turbine. So for the same cycle and airflow it may be advantageous to have higher speed. This may be obtained by reduction of flow capability of the compressor at given speed. This leads to a rematching study of the first stages.

That discussion confirms the interest of advanced research towards highly variable geometry engines.

6.2.3. The status engine model is used first to calculate derivatives or influence coefficients of the different factors listed in the above table. These derivatives have to be calculated for the important operating points. Then it is possible to appreciate what modification have the best pay-off capability; more detailed studies and simulations will be conducted for the selected ones. Simulation of component modifications may be carried out by G factors changes.

6.3. Control schedules optimization

While the engine and its components are analyzed and developed, progress is also going on for systems and particularly the control system. Requirements and specifications of it have to be revised and up-dated to fit real characteristics of engine and components.

Choice of control parameters and definition of schedules have to be studied again every time that a significant change has been recognized in the engine characteristics.

Many control system functions are applicable to transient operations. So transient and dynamic simulation of the engine and its associated control system are of major interest. Versatility of digital math models yields the opportunity to evaluate effects of possible components change on control schedule adaptability.

In that type of study, quality of control system modelling is of prime importance: carefull integrated work with control designers is required to add the control system digital model to engine model program.

Control schedules have to be defined in order to optimize performance and stability particularly for the installed engine. Then air intake system and after body-exhaust nozzle models must be added. Also it may be very useful to have identified by engine test, correction factors for distorsion effects on components characteristics; thus it will be possible to account for this effect and predict performance plus optimize the control schedule provided that the distorsion induced by the inlet system is known. Of course analysis of flight test and simulated altitude test with the current control schedules is very important to clearly establish the validity of the model before using it to optimize and redefine control schedules and system, particularly because a number of control system characteristics are unaccurately known or predicted.

6.4. Component change evaluation

Beyond minor adjustments to rematch the engine it may be necessary to consider extensive change in the components either for mechanical reasons or because rematching appears to be insufficient to reach the specified performances. Also engine growth studies require evaluation of new component design.

Direction of new design requirements is shown by influence studies. Component designer will produce prediction for possible new concepts, taking advantage of on-engine characterization established for the former component.

Thanks to its modular structure, input of that characteristic in the model is easy and extensive study of the possible performance can be quite rapidly performed. Results will be a precious guide to decide an effective change of the component as it is generally a costly and time consuming operation. Simulation of results to be expected from tests will help to organize it and to check rapidly if the predicted performance is obtained.

6.5. Fault isolation

At a time when maintenance problems and costs become more and more critical with sophisticated modern engines, simulation techniques can contribute efficiently to identify faulty operation.

As a matter of fact a modular model, particularly when built with correction factors capability, gives the possibility of component deterioration simulation.

Deterioration of a component may be simulated by change in its characteristics and mainly flow capacity and efficiency level in its map.

As for test analysis method described in 5, identification of G Factors measuring the deterioration effects can be performed by simulation of its consequences on measured parameters.

Up to now, for practical reasons, diagnostic techniques have been based on linear models built around a matrix of influence factors calculated with the complete non-linear model for a current operational point.

That matrix is utilized to calculate relative changes in component characteristics i.e G factors variations, which simulate with the best accuracy, deviation of measured parameters from reference normal

values. Linear regression method is a valuable way to derive the G factors vector representing the most probable solution.

7. CONCLUSION AND RECOMMENDATIONS

Clear knowledge of component characteristics as they work on the engine requires extensive analysis of test results. It is important to derive from the measurements a view consistent with the scheme of representation used in design and prediction. However too rigid modelling may hide unpredicted phenomenons as it leads to the classic trap of fitting reality to the model. It is necessary to carefully coordinate evolution of modelling techniques with prediction and test analysis methods.

Fair balance between test quantity and analysis capability must be managed to efficiently develop engine performance as none of each can be used alone to give the right information. Methods of analysis described are quite powerful and may give the capability of extracting maximum information from often expensive tests. However it requires substantial manpower and computer means to give valuable payoff. More sophisticated and expensive future engines will undoubtly emphasize investments in prediction, test analysis and simulation methods.

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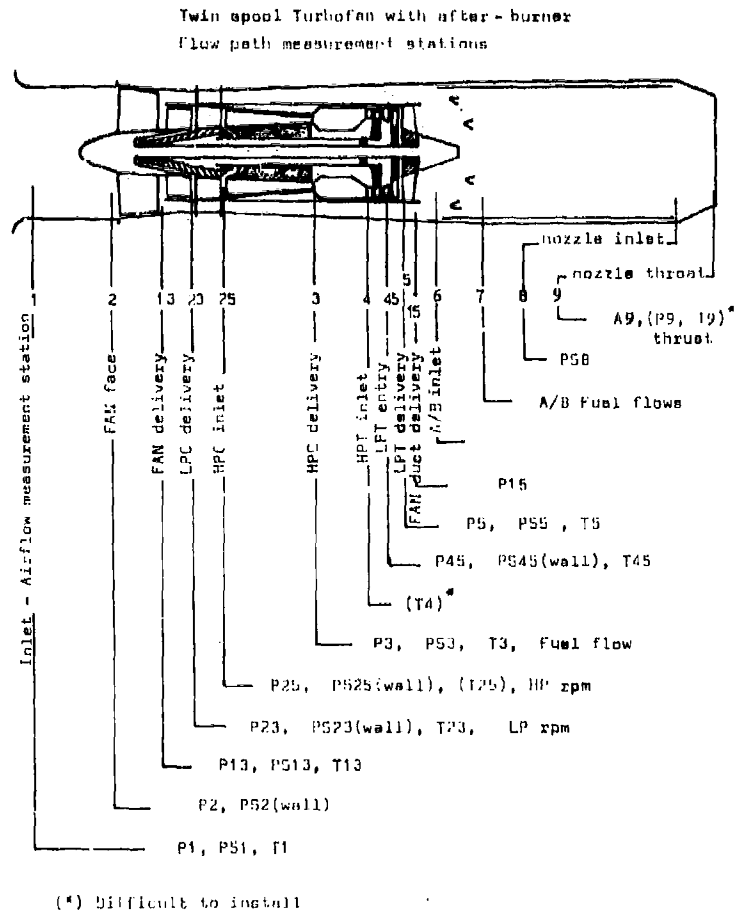

Figure 1

METHOD	MAIN CALCULATION PATHS			
	Measurements	Estimated data	Equations	Derived parameter
1.- HPT flow function	P3, (P45) T3 WF T2, T13 W2	- HPT flow function - Burner pressure loss - Burner efficiency - Compressor bleeds	HPT continuity Combustion heating Energy balance	- Primary airflow - Turbine entry Temperature T4 - T45, T5
2.- Energy balance	T2,T3,T13,T5 W2 WF p3 T25	- Burner efficiency - Bleeds - Burner pressure loss	Energy balance Combustion heating	- Primary airflow - Turbine entry Temperature T4 - HPT flow function - LPT entry temp. T45
3.- LPT flow function	P45, P5 T25 WF P3, T3	- LPT flow function - Burner efficiency - Bleeds	LPT continuity Combustion heating Energy balance on HP spool	- Primary airflow - LPT entry temperature - HPT entry temperature - HPT flow function
4.- FAN duct flow function	P13,PS13,T13 W2 WF P3, T3 T2, T25	- FAN duct effective aera - Combustion efficiency - Bleeds	Continuity Combustion heating Energy balance	- Secondary airflow - Primary airflow - T entry temperature - LPT entry temp. T45 - HPT and LPT flows functions - T5

Figure 2

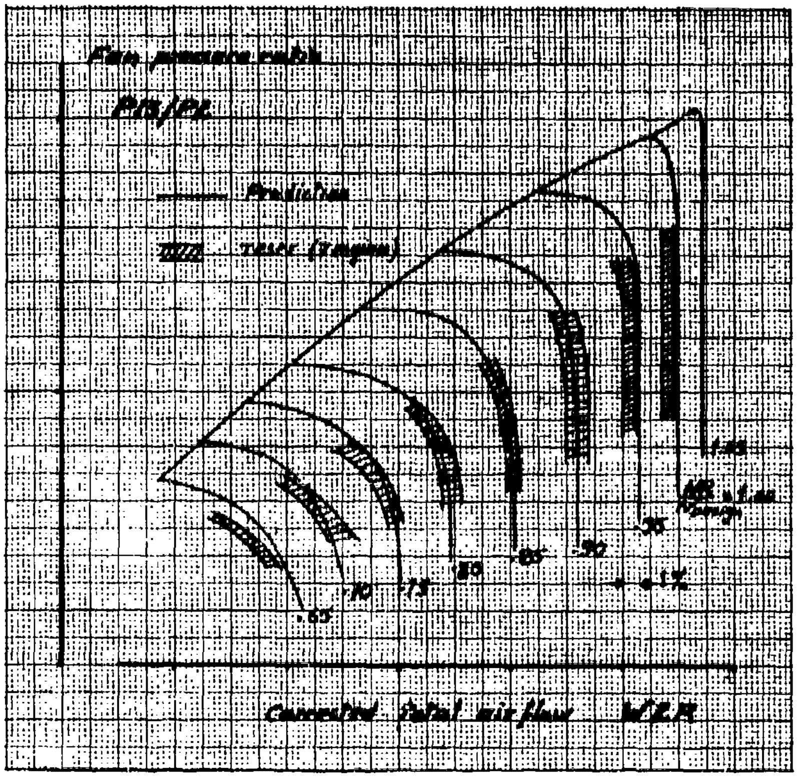

Figure 3

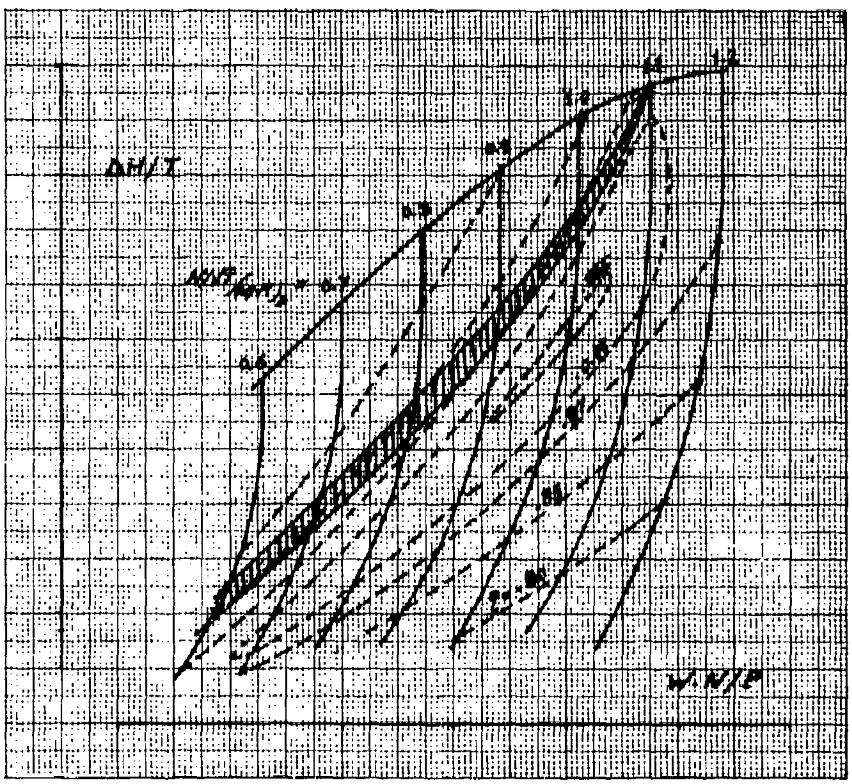

Figure 4

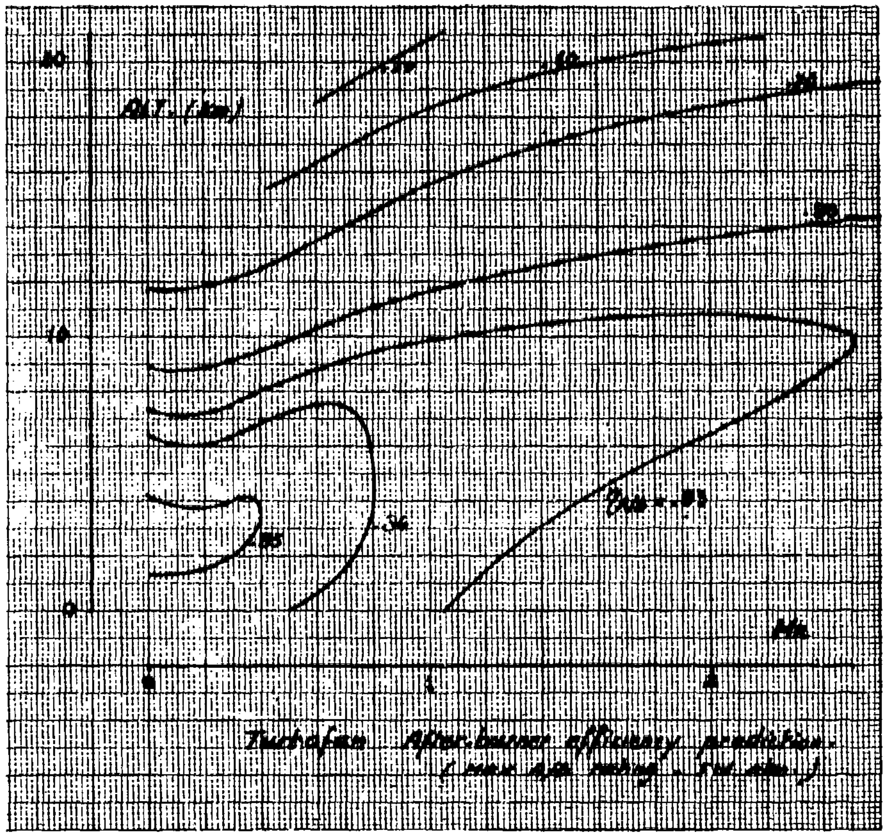

Figure 5

COMPRESSOR MAP TRANSFORMATION	POSSIBLE PHYSICAL CORRELATION	CORRECTION FACTOR RELATION (see définition page)
- Scale flow and efficiency at constant corrected speed and pressure ratio	Change in géometrical size	g3 = 1, g2.g4 = 1, g1, g2
- Scale flow and efficiency at constant corrected speed and ΔH/T	Losses change	g3 = 1, g4 = 1, g1, g2
- Scale flow and efficiency at constant corrected speed and throttling	Change in stage stagger	g3 = 1, g2.g4 = f (g1, pressure ratio) (simplified : g2.g4 = g1)
- Scale flow and efficiency at constant corrected speed and throttling		g1 = 1, g4 = 1, g3, g2
- Scale flow and pressure ratio at constant corrected speed and efficiency	Change in stage deviation	g3 = 1, g2 = 1, g1, g4
	Change in circumferential speed	g2 = 1, g4 = g3, g1, g3
TURBINE MAP TRANSFORMATION		
- Scale flow and efficiency at constant corrected speed and ΔH/T	Losses change	g3 = 1, g4 = g1, g1, g2
- Scale flow and efficiency at constant corrected speed and pressure ratio	Change in stage stagger	g3 = 1, g4 = g1.g2, g1, g2

Figure 6

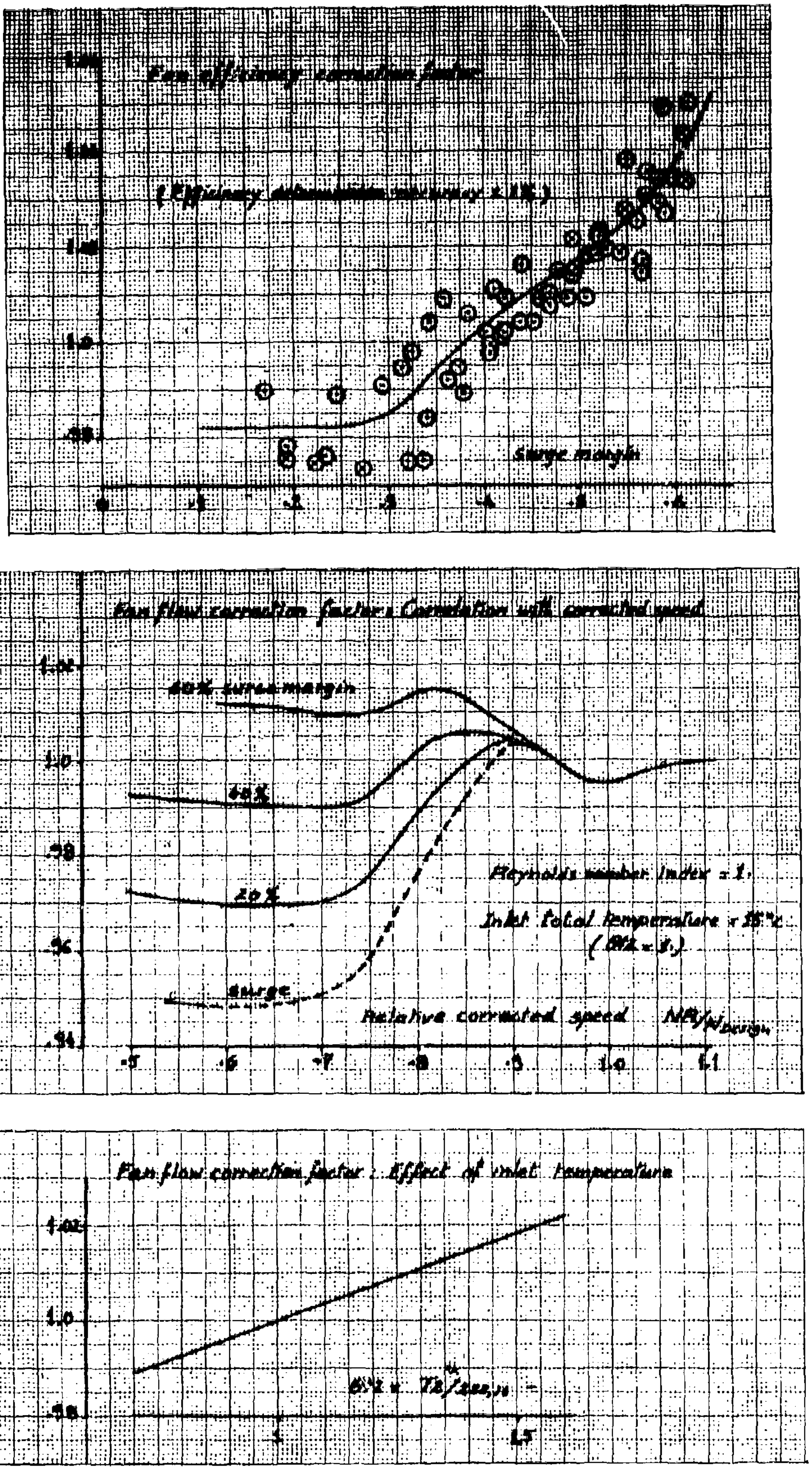
Fan efficiency correction factor
Surge margin
1.0
Fan flow correction factor : Correlation with corrected speed
40% surge margin
40%
20%
surge
Reynolds number Index = 1.
Inlet total temperature = 15°C
Relative corrected speed
1.0
.98
.96
.94
1.0
1.1
Fan flow correction factor : Effect of inlet temperature
1.02
1.0
.98
1.5

Figure 7

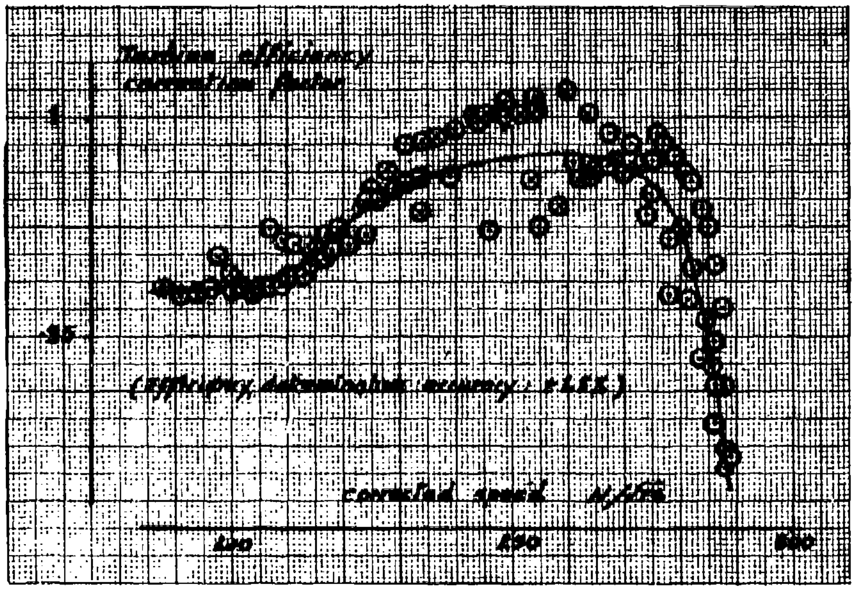

Figure 8

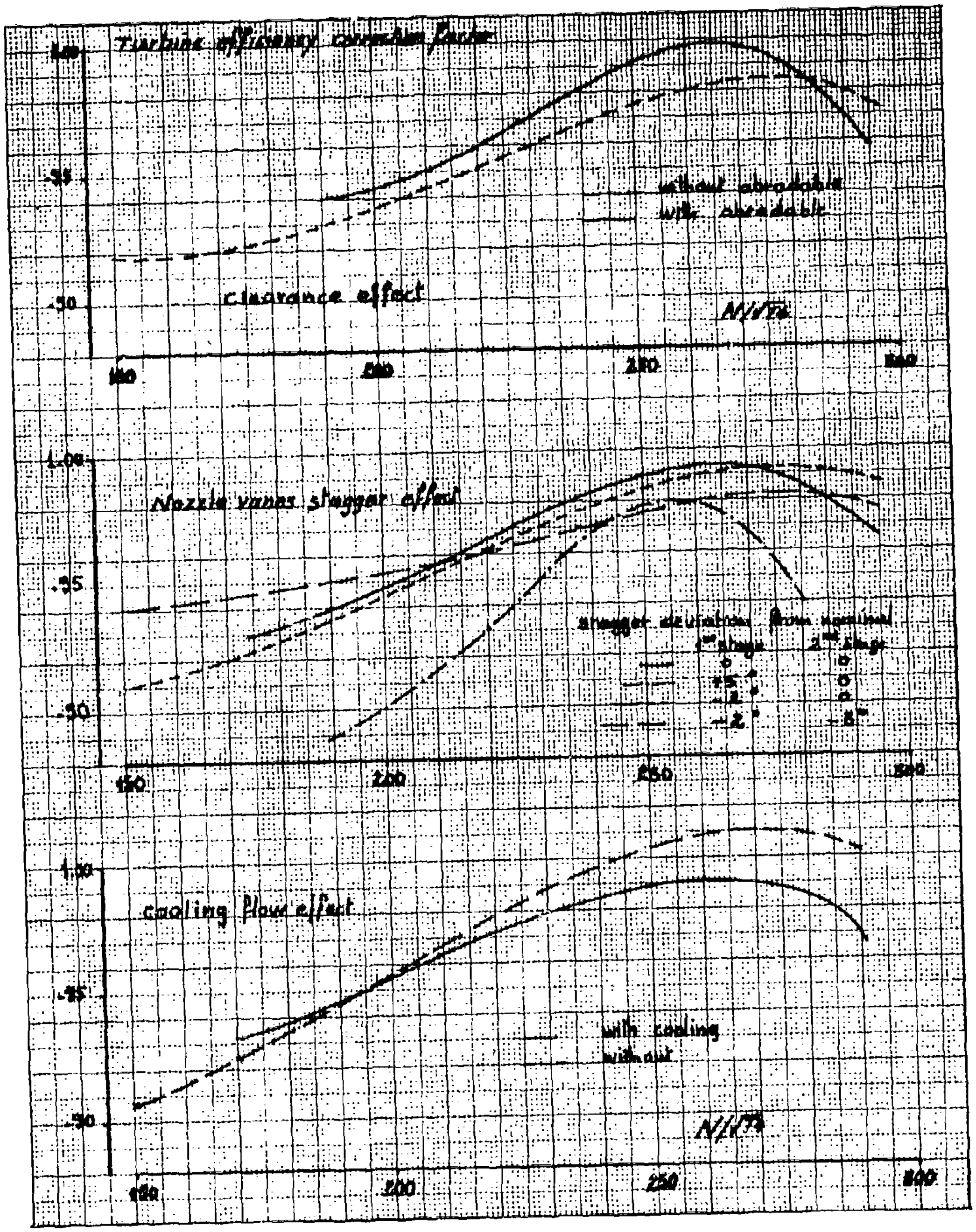
Turbine efficiency correction factor
.95
.90
without abradable
with abradable
clearance effect
N/√Tt
250
1.00
Nozzle vanes stagger effect
.95
stagger deviation from nominal
1st stage
2nd stage
0
+5
-2
-8
.90
150
200
250
300
1.00
cooling flow effect
.95
with cooling
without
.90
N/√Tt
150
200
250
300

Figure 9

B

BIBLIOGRAPHY

on

MODERN PREDICTION METHODS FOR TURBOMACHINE PERFORMANCE

prepared by

A S Reeves
Defence Scientific Information Service
Department of National Defence
Ottawa, Ontario

AGARD, the Lecture Series Director and the speakers acknowledge the valuable contribution to the Lecture Series by Mr A S Reeves of the Canadian Defence Scientific Information Service in compiling the Bibliography at the request of the AGARD Technical Information Panel and the Director of Plans & Programmes.

HANDBOOK, UNCERTAINTY IN GAS TURBINE MEASUREMENTS.
Abernathy, R. B.; Powell, B. D.; Colbert, D. L.; Sanders, D. G.; Thompson, J. W. Jr.
Arnold Engineering Development Center Arnold Air Force Station Tenn.
Feb 73; 181p
AEDC-Tr-73-5
AD-755 356
The lack of a standard method for estimating the errors associated with gas turbine performance data has made it impossible to compare measurement systems between facilities, and there has been confusion over the interpretation of error analysis. Therefore, a standard uncertainty methodology is proposed in this Handbook. The mathematical uncertainty model presented is based on two components of measurement error: the fixed (bias) error and the random (precision) error. The result of applying the model is an estimate of the error in the measured performance parameter. The uncertainty estimate is the interval about the measurement which is expected to encompass the true value. The propagation of error from basic measurements through calculated performance parameters is presented. Traceability of measurement back to the National Bureau of Standards and associated error sources is reviewed. (author)

THREE - DIMENSIONAL TURBULENT BOUNDARY LAYER IN A ROTATING HELICAL CHANNEL.
Anand, A. K.; Lakshminarayana, B.
PA State Univ., University Park.
ASME Pap. N 74-FE-25 for Meet. May 13-15 1974, p.16

TURBINE ENGINE CONTROL SYNTHESIS.
Arnett, S. E.
Bendix Corp South Bend Ind Energy Controls Div Air Force Aero Propulsion Lab. Wright-Patterson AFB, Ohio.
Dec 74; 212p.
ECD-863-18667-R
AD-A-005 817
A highly versatile, research type, externally programmable control system for use in development and evaluation of new modes of control was assembled at the Air Force Aero-Propulsion Laboratory. It includes: sensors and transducers to measure and transform engine operating state, engine geometry actuation devices and fuel control mechanisms, a J85-13 engine mounted the cell of Room 21, Building 18C of the AFAPL, a simulation of the J85 engine on the AFAPL's Applied Dynamics AD/Five Computer, programs for the IBM 1800 computer to control the engine and the simulated engines, and interface electronic equipment to complete the circuit between the digital computer, engine-mounted equipment and the simulated engine.

PERFORMANCE OF THREE VANED RADIAL DIFFUSERS WITH SWIRLING TRANSONIC FLOW.
Baghdadi, S.; McDonald, A. T.
ASME Pap. N 75-FE-19 for Meet. May 5-7 1975, p.8

PERFORMANCE OF 1380 FOOT PER SECOND TIP-SPEED AXIAL-FLOW COMPRESSOR ROTOR BLADE TIP SOLIDITY OF 1.5.
Ball, C. L.; Janetzke, D. C.; Reid, L.
National Aeronautics and Space Administration. Lewis Research Center, Cleveland, Ohio.
Jan. 1972 100p. refs.
NASA-TM-X-2379; E-5723
N72-14989
This presents the aerodynamic design parameters along with the overall and blade element performance of an axial-flow compressor rotor designed to study the effects of blade solidity on efficiency and stall margin. At design speed the peak efficiency was 0.892 and occurred at an equivalent weight flow of 65.0 lb/sec. The total pressure ratio was 1.83 and the total temperature ratio was 1.215. Design efficiency, weight flow, pressure ratio, and temperature ratio were 0.824, 65.3, 1.65, and 1.187, respectively. Stall margin for design speed was 10 percent based on the weight flow and pressure ratio values at peak efficiency and just prior to stall. (author)

SIMULATION OF AXIAL COMPRESSOR PERFORMANCE USING AN ANNULUS WALL BOUNDARY LAYER THEORY.
Balsa, T. F.; Mellor, G. L.
Dynalysis of Princeton, NJ.
ASME Pap. N 74-GT-56 for Meet. Mar. 31-Apr. 4 1974, p.13 ASMSA

TURBINE ENGINE TRANSIENT OPERATION.
Barbot, A.
SNECMA, Fr
Entropie V 9. N 51. May-Jun 1973, p. 43-47

SECONDARY FLOW EFFECTS ON GAS EXIT ANGLES IN RECTILINEAR CASCADES.
Bardon, M. F.; Moffatt, W. C.; Randall, J. L.
Can. Armed Forces, Calgary, Alta.
J Eng Power Trans ASME V. 97 SER A N 1 Jan 1975 P.93-100

BOUNDARY-LAYER PREDICTION METHODS APPLIED TO COOLING PROBLEMS IN THE GAS TURBINE.
Bayley, F. J.; Morris, W. D.; Owen, J. M.; Turner, A. B.
Sussex Univ., Brighton (England). Lab for Mechanical Engineering.
London Aeron. Res. Council 1971 43p. refs.
ARC-CP-1164; ARC-32050
N72-11300
Integral and differential theories have been used, and applied to flows over convection- and transpiration-cooled turbine blades and simplified representations of combustion systems and turbine disks.

HEAT TRANSFER ANALYSIS ALONG THE BLADES OF A GAS TURBINE STATOR BY THERMAL AND KINEMATIC BOUNDARY LAYERS THEORY
Becko, Y.
ACEC - Westinghouse Res. Lab., Charleroi, Belg.
ASME Pap. N 75-GT-15 for Meet. Mar 2-6 1975, p.8

COLD-AIR INVESTIGATION OF A TURBINE FOR HIGH TEMPERATURE-ENGINE APPLICATION. 5: TWO-STAGE TURBINE PERFORMANCE AS AFFECTED BY VARIABLE STATOR AREA.
Behning, F. P.; Schum, H. J.; Szanca, E. M.
National Aeronautics and Space Administration. Lewis Research Center, Cleveland, Ohio.
Apr. 1974, 40p.
NASA-TN-D-7571; E-7671
N74-19642
The stator areas of the design two-stage turbine were both decreased and increased by nominally 30 percent, and the performances of the two turbines are compared with that of the design stator area turbine. Turbine efficiency decreased with stator area changes. Closing the stator area resulted in the more severe efficiency loss. The decrease in efficiency for both turbines is attributable to rotor incidence, off-design blade-surface velocities, and adverse reaction changes across the blade rows.

EXPERIMENTAL STUDY OF ROTATING STALL IN HIGH-PRESSURE STAGES OF AN AXIAL FLOW COMPRESSOR.
Beknev, V. S.; Zemlyanskiy, A. V.; Tumashev, R. Z.
Techtran Corp., Glen Burnie, Md.
Sep. 1973, 10p.
NASA-TT-F-15115
N73-31698
An experimental study of rotating stall in axial-flow compressor stages with different types of profiling along the blade height and with different calculated regimes of flow past a profile in the cascade was conducted. It is found that, in spite of the different safety margins with respect to boundary layer separation in the cascades of the different stages, their boundaries of stable operation are almost the same. It is shown that profiling taking into account end effects has a stabilizing influence and leads to a smoother transition to the rotating stall regime. The flow in rotating stall zones is shown to be of three-dimensional nature. It is shown that this three-dimensional structure can be detected with the aid of straight and L-shaped tensoanemometer probes.

ANALYTICAL PREDICTION OF FAN/COMPRESSOR NOISE.
Benzakein, M. J.; Morgan, W. R.
General Electric Co., Cincinnati, Ohio.
ASME Pap. 69-WA/GT-10 for Meet. Nov 16-20 1969, p.8

CALCULATION OF THE QUASI-THREE-DIMENSIONAL FLOW IN AN AXIAL GAS TURBINE.
Biniaris, G.
Rhein-Westf Tech. Hochsch, Aachen, Ger.
ASME Pap. N 74-GT-67 for Meet. Mar 31-Apr 4 1974, p.12

NEW TECHNIQUES FOR MEASURING FILM COOLING EFFECTIVENESS AND HEAT TRANSFER.
Blair, M. F.; Lander, R. D.
ASME Pap. N 74-HT-8 for Meet. Jul 15-17 1974, p.8

INFLUENCE OF COOLING, FREE-STREAM TURBULENCE AND SURFACE-ROUGHNESS ON THE AERODYNAMIC BEHAVIOR OF CASCADES.
Boekenbrink, D.
Tech. Hochsch, Aachen, Ger.
ASME Pap. N 74-WA/GT-9 for Meet. Nov 17-22 1974, p.16

AIRCRAFT ENGINES (SELECTED ARTICLES)
Bordovitzyn, Yu. A.; Kalmykov, I. I ; Strunkin, V. A.; Dyatlov, I. N.;Sharapov, A. V.
Foreign Technology Div. Wright-Patterson AFB, Ohio.
7 May 71, 86p.
FTD-MT-24-18-71
AD-727 175
Contents: The effect of the form of an active gas jet on the characteristics of a gas ejector with short mixing chamber: Calculation of the maximum degree of compression of an ejector: The rational method of selecting parameters and calculating single-stage gas turbines: A generalization of the results of measurements of the fineness of fuel atomization of mechanical and air-mechanical injectors of the pressure-jet type: Calculation of temperature field in a cooled turbine blade with longitudinal cooling channels: Turbulence in the zone of interplay of jetstreams with the flow.

IMPROVED METHOD FOR CALCULATING THE FLOW IN TURBO-MACHINES, INCLUDING A CONSISTENT LOSS MODEL.
Bosman, C.; Marsh, H.
Univ. of Manchester, Engl.
J Mech. Eng. Sci. Vol. 16, No. 1, Feb 1974, p.25-31

THE EFFECT OF DAMPING FACTOR ON THE BEHAVIOUR OF FLOW CALCULATIONS IN TURBOMACHINES.
Bosman, C.
Manchester Univ. (England). Dept. of Mechanical Engineering.
1975, 35p.
ARC-R/M-3766, ARC-35099
N75-28034

SUB-OPTIMAL CONTROL OF A GAS TURBINE ENGINE.
Bowles, R. J.
Air Force Inst. of Tech. Wright-Patterson AFB, Ohio School of Engineering.
Dec 73, 126p.
GE/EE/73A-1
AD-777 852
Gas turbine engine technology has been advanced to the point where it is increasingly difficult to apply conventional feedback control theory to the design of its necessary multiple-input and multiple-output control systems. Modern linear control theory was applied to the engine control problem in four stages. Stage 1 involved formulating a simple, reliable, and representative mathematical model of the engine and describing

the model in state variable form. Stage 2 involved using a truncated Taylor series to linearize the engine state equations about a succession of engine equilibrium points and solving the linear optimal output regulator problem and the linear optimal tracking problem to minimize a quadratic Performance Index. In stage 3, the engine model was simulated on a digital computer via the Mimic simulation Language. The engine control laws were then realized in a Fortran function sub-program. Stage 4 consisted of testing the controlled engine for various throttle demands to verify the satisfactory operation of the controller.

SMALL AXIAL-CENTRIFUGAL COMPRESSOR MATCHING STUDY.
Brown, L. E.
Curtiss-Wright Corp., Wood-Ridge, N.J.
May 70, 224p.
USAAVLABS-TR-70-20
AD-873 844
The prime objective of the program is to define the preliminary design and matching of an axial-centrifugal compressor for minimum engine specific fuel consumption at 60 percent power and 30 percent power, with secondary importance attached to SFC at 100 percent power. Analytical procedures were employed to investigate effects of engine cycle parameters, stage-matching characteristics of several axial and centrifugal compressors, variable compressor geometry (two-spools and stator variable setting angles), and power turbine variable area, upon minimum fuel consumption. Design studies were employed in the consideration of shafting and component arrangements. Comparative engine complexity of one-spool and two-spool compressors in front drive engines was evaluated in the selection of an optimum compressor configuration.

STEADY- STATE CIRCUMFERENTIAL INLET PRESSURE DISTORTION INDEX FOR AXIAL - FLOW COMPRESSORS.
Brunda, D. F.; Boytos, J. F.
Proc. 10th Nat. Conf. on Environmental Effects on Aircraft and Propulsion Systems, Trenton-Philadelphia, May 18-20 1971,

MINIATURIZED HIGH SPEED CONTROLS FOR TURBINE ENGINES (FABRICATION AND TEST)
Burnell, D. G.; Cole, M. A.; Morrison, T. B.; White, A. H.; Zagranski, R. D.
Colt Industries Inc. West Hartford Conn. Chandler Evans Control Systems Div. Air Force Aero Propulsion Lab., Wright-Patterson AFB, Ohio.
Aug 74, 205p.
AFAPL-TR-74-93
AD-A-006 108
This report summarizes the design and development of control components and high speed fuel pump technology for future drone, missile and RPV gas turbine engines. The hardware was designed for installation on an existing engine to provide a vehicle for control mode demonstration and evaluation testing. The developed hardware includes a fluidic/linear variable differential transformer delta P/P airflow sensor, a radiation pyrometer for sensing turbine blade temperature, three pressure transducers for sensing burner pressure (variable capacitance, strain gage, and potentiometric types), a 70,000 rpm centrifugal pump and proportional solenoid-operated fuel metering system, a magnetic clutch-driven 12 rpm gear pump for fuel metering and fuel pressurization, and an electronic unit for interfacing these components with an engine test cell digital computer.

PERFORMANCE PARAMETERS FOR GAS GENERATOR COMPARISON.
Burns, R. J.
Naval Postgraduate School Monterey Calif.
Mar 72, 86p.
AD-742 940
Methods for comparing gas turbine engines where the thermodynamic cycle begins and ends in the atmosphere are well defined and documented. No such comparison technique(s) exists for the gas generator or core portion of the engine. The term gas generator or core refers to the high pressure compressor and turbine, and the combustor. The thesis formulates gas generator performance parameters, develops methods of testing and data reduction necessary to obtain these parameters, establishes criteria for comparing two gas generators, and develops an analytical model to test the validity of the comparison method.

INVESTIGATION OF FACTORS AFFECTING SMALL TURBINE EFFICIENCY AND LOSS PREDICTION.
Burrows, L. T.
Army Aviation Materiel Labs., Fort Eustis. Va.
Jun 69, 43p.
USAAVLABS-TR-69-54
AD-859 273
The aerodynamics and mechanics of the high-temperature, low-mass-flow, low-aspect ratio, axial-flow turbine are discussed. The problems of small size and resulting high secondary losses are considered, and approaches to improving efficiency are offered, in addition, experimental analysis of a turbine stator annulus (aspect ratio = 0.5) is presented, with recommendation for an accurate prediction of the losses in the small, high-performance turbine.

INSTANTANEOUS DISTORTION IN A MACH 2.5, 40-PERCENT-INTERNAL-CONTRACTION INLET AND ITS EFFECT ON TURBOJET STALL MARGIN.
Burstadt, P. L.; Calogeras, J. E.
Jul 1974, 42p.
NASA-TM-X-3002; E-6294
N74-30243
An experimental wind tunnel investigation was conducted to determine the effects of time-variant distortions produced in a supersonic inlet on a J85-GE-13 turbojet engine. Results are presented principally in terms of instantaneous distortion amplitudes and total-pressure contours measured through compressor stall. They indicate that, although a time-averaged distortion may be far from a stall-inducing value, corresponding instantaneous distortion amplitudes can approach or exceed this value. Seven engine stall events were studied. In six of these events instantaneous distortion peaks of sufficient magnitude to cause stall were observed in the time period shortly before stall was detected.

CENTRIFUGAL COMPRESSOR ANTI-SURGE AND RECYCLE CONTROL.
Buzzard, W. S.
Adv in Instrum V 28, Annu ISA Conf, 28th, Proc, Houston, Tex,
Oct 15-18 1973, Part 2, Pap 610, p.8

DESIGN AND EXPERIMENTAL EVALUATION OF A HIGH-TEMPERATURE RADIAL TURBINE, PHASE 2
Calvert, G. S.; Bock, S. C.; Okapuu, U.
Army Air Mobility Research and Development Lab. Ft. Eustis, Va.
May 1971, 223p.
PWA-FR-4058
N71-37381
The report describes the design, fabrication and test of a radial turbine designed to produce 219.6 Btu/lb stage work at 87.5% efficiency, with a 5:1 stage pressure ratio. Turbine inlet gas conditions at design point were 257.5 psia and 2300 F. The resulting turbine configuration consisted of an air-cooled, 12-bladed rotor designed for 67,000 rpm, and a 20-vaned air-cooled nozzle section of a reflex-type (supersonic) design. Both parts were designed as IN100 (PWA 658) investment castings. As part of the preliminary design effort, a fabrication study was conducted to evaluate feasible methods of casting the turbine nozzle and rotor. Results showed that the nozzle section could be cast as an integral assembly, but fabrication of the rotor as an integral casting was much more difficult. Bicasting was evaluated as an alternate method of fabricating the rotor, and results showed substantial advantages for the bicasting technique. However, neither method could produce designed rotor properties, and testing was conducted with structurally limited rotors. A test rig was designed and fabricated by the contractor. The test rig consisted of a supercharged gas generator, which had the capability of controlling the turbine load by varying the compressor flow rate. Burner testing preceded turbine testing.

TEST RESULTS OF A VTOL PROPULSION CONCEPT UTILIZING A TURBOFAN POWERED AUGMENTOR
Campbell, D. R.; Quinn, B.
Aerospace Research Labs Wright-Patterson AFB Ohio
3 May 74, 7p,
ARL-75-0046
AD-A007-757
Several questions relevant to the feasibility of achieving successful VTOL flight with thrust agumenting ejector wings are answered by the present experimental study. Tests were performed with a large-scale turbofan powered augmentor that embodied many of the problems encountered in the design of real flight hardware. The apparatus consisted of four separate and parallel ejector channels in each of two wings. Results were compared with data from other laboratory experiments using a single-channel ejector of similar geometry. Over-all performance levels of the large multichannel apparatus correlated well with the single-channel results. Minor interactions between the four ejector channels on each wing had no significant effect on the over-all level of thrust augmentation. However, the distribution of thrust was affected and should be considered in future aircraft system designs. Operating as an air pump, the turbofan engine was maintained in its safe operating regime throughout all test configurations.

ENGINE CONTROL SYSTEMS STUDY AS APPLIED TO INTER-ENGINE THRUST CONTROL
Carras, A. N.; Hughett, P. W.
Inter-Controls Inc. Washington D.C.
Jan 70, p.246

AD-709 411
VTOL type aircraft incorporating turbo-fan engines as lifting means do not sensibly lend themselves to the cross-coupling provisions inherently available with the shafting of propeller type engines. An engine failure in the fan engine case is therefore a considerably more precarious matter for which provision for thrust compensation more responsive than a pilot would appear to be required. The study utilizes a very comprehensive hybrid simulation of the Wtf-60 engine wherein all engine components are simulated on a performance map basis thereby including all non-linearities as well as permitting the availability of any and all engine parameters for use as controlled variables operating in conjunction with the manipulated variable, fuel flow. Further, realistic acceleration control in the course of large upsets is accomplished thereby permitting a control system analysis which is completely applicable to the detail design of the control system and the selection of components.

ANALYSIS OF UNSTEADY AERODYNAMIC EFFECTS ON AN AXIAL-FLOW COMPRESSOR STAGE WITH DISTORTED INFLOW.
Carta, F. O.
Purdue Univ. Lafayette Ind Project Squid Headquarters
Jul 72, 124p.
SQUID-TR-UARL-1-PU
AD-746 457
An analytical procedure has been developed to predict the circumferential pressure profile at the exit of a compressor stage subjected to a spatially steady inlet distortion. Expressions have been developed relating the pressure ratio and weight flow characteristics of an axial-flow compressor stage to the normal force and drag characteristics of an isolated airfoil. These steady-state interrelations between rotor and isolated airfoil were used to apply available unsteady data for isolated airfoils in the study of distorted inflow effects. Both unsteady and quasi-steady predictions were made and were compared with available experimental results from low-speed compressor tests.

ANALYSIS OF GEOMETRY AND DESIGN-POINT PERFORMANCE OF AXIAL-FLOW TURBINES USING SPECIFIED MERIDIONAL VELOCITY GRADIENTS.
Carter, A. F.; Lenherr, F. K.
Northern Research and Engineering Corp., Cambridge, Mass.
Dec 69, 240p.
NASA-CR-1456; NREC-1147-1
N70-14335

STRESS CALCULATIONS FOR LIFETIME PREDICTION IN TURBINE BLADES.
Chaboche, J. L.
Off Natl D'Etud et de Rech Aerosp, Chatillon, Fr Int J Solids Struct
V 10, N 5, May 1974, p.473-482

GAS TURBINE CYCLE CALCULATIONS: EXPERIMENTAL VERIFICATION OF OFF-DESIGN-POINT PERFORMANCE PREDICTIONS FOR A TWO-SPOOL TURBOJET WITH VARIOUS AIR BLEEDS.
Chappell, M. S.; Grabe, W.
National Research Council of Canada, Ottawa (Ontario) Div. of Mechanical Engineering.
Nov 1971, 76p. refs
LR-555; NRC-12475
N72-22798
A simplified method for calculating off-design point performance of turbojet and turbofan engines, both at sea level and at altitude conditions, is presented. The method implies constancy of component efficiencies and linearity of corrected mass flow with corrected engine speed. Data were gathered on a J-75 two spool turbojet engine at part throttle conditions, with compressor bleed extraction, and with propelling nozzle area change. The calculations were found to be accurate for part throttle performance, but less successful for the other conditions.

SIMILARITY CONSTRAINTS IN TESTING OF COOLED ENGINE PARTS.
Colladay, R. S.; Stepka, F. S.
National Aeronautics and Space Administration. Lewis Research Center. Cleveland, Ohio.
Jun 74, 23p.
NASA-TN-D-7707
N74-27429
A study is made of the effect of testing cooled parts of current and advanced gas turbine engines at the reduced temperature and pressure conditions which maintain similarity with the engine environment. Some of the problems facing the experimentalist in evaluating heat transfer and aerodynamic performance when hardware is tested at conditions other than the actual engine environment are considered. Low temperature and pressure test environments can simulate the performance of actual size prototype engine hardware within the tolerance of experimental accuracy if appropriate similarity conditions are satisfied. Failure to adhere to these similarity constraints because of test facility limitations or other reasons, can result in a number of serious errors in projecting the performance of test hardware to engine conditions.

STABILITY OF AN AXIAL FLOW COMPRESSOR WITH STEADY INLET CONDITIONS.
Corbett, A. G.; Elder, R. L.
J Mech Eng Sci V. 16, N 6, Dec 1974, p.377-385

AN EXPERIMENTAL INVESTIGATION OF COMPRESSOR STALL USING AN ON-LINE DISTORTION INDICATOR AND SIGNAL CONDITIONER
Costakis, W. G.; Wenzel, L. M.
National Aeronautics and Space Administration. Lewis Research Center, Cleveland, Ohio.
Apr 1975, 33p. refs.
NASA-TM-X-3182; E-8120
N75-20248
The relation of the steady-state and dynamic distortions and the stall margin of a J85-13 turbojet engine was investigated. A distortion indicator capable of computing two distortion indices was used. A special purpose signal conditioner was also used as an interface between transducer signals and distortion indicator. A good correlation of steady-state distortion and stall margin was established. The prediction of stall by using the indices as instantaneous distortion indicators was not successful. A sensitivity factor that related the loss of stall margin to the turbulence level was found.

TOWARDS THE EFFICIENT VIBRATION ANALYSIS OF SHROUDED BLADED DISK ASSEMBLIES.
Cottney, D. J.; Ewins, D. J.
J Eng Ind, Trans ASME V. 96, Ser B N 3, Aug 1974, p.1054-1059

COMPUTER PROGRAM FOR DEFINITION OF TRANSONIC AXIAL-FLOW COMPRESSOR BLADE ROWS
Crouse, J. E.
National Aeronautics and Space Administration. Lewis Research Center, Cleveland, Ohio.
Feb 1974, 224p. refs.
NASA-TN-D-7345; E-7094
N74-17701
A method is presented for designing axial-flow compressor blading from blade elements defined on cones which pass through the blade-edge streamline locations. Each blade-element centerline is composed of two segments which are tangent to each other. The centerline and surfaces of each segment have constant change of angle with path distance. The stacking line for the blade elements can be leaned in both the axial and tangential directions. The output of the computer program gives coordinates for fabrication and properties for aeroelastic analysis for planar blade sections. These coordinates and properties are obtained by interpolation across conical blade elements. The program is structured to be coupled with an aerodynamic design program.

A METHOD FOR EVALUATING OFF-DESIGN PERFORMANCE OF A RADIAL INFLOW TURBINE & COMPARISON WITH EXPERIMENTS.
Dadone, A.; Pandolfi, M.
Politecnico Di Torino (Italy) Ist. Di Macchine E Motori Per Aeromobili.
Aug 68, 25p.
N70-23653

THE OFF-DESIGN ANALYSIS OF FLOW IN AXIAL COMPRESSORS.
Daneshyar, H.; Shaalan, M. R. A.
Cambridge Univ. (England) Dept. of Engineering.
1972, 51p. refs.

ARC-CP-1234; ARC-32727
N73-18293
The existence and uniqueness of the solutions obtained from the streamline curvature method of calculating flow through turbomachines are examined for several operating points of Rolls-Royce compressors. It is shown that under certain conditions the truncation errors in the numerical solution can become large and hence give rise to the violation of the uniqueness conditions. The computer program may then give wrong answers to the physical problem. The conditions for existence and uniqueness may be violated when the meridional velocities are small (e.g., near stall) or when there are regions of choked flow. Flow for an operating point in the stall region is computed by suitable modifications to minimize the truncation errors and hence to obtain a unique solution. This is compared with the results of the previously reported actuator disk theory and experiment. The effect of variation of losses on the calculation is examined together with the effect of a correction term due to a dissipative body force, which should be included in the momentum equation when losses are introduced.

PREDICTION OF AXIAL-FLOW INSTABILITIES IN A TURBOJET ENGINE BY USE OF A MULTISTAGE COMPRESSOR SIMULATION ON THE DIGITAL COMPUTER
Daniele, C. J.; Blaha, R. J.; Seldner, K.
National Aeronautics and Space Administration. Lewis Research Center, Cleveland, Ohio.
Jan 75, 36p.
NASA-TM-X-3134; E-8008
N75-13870
A method of estimating the undistorted stall line for an axial-flow compressor by using the digital computer is presented. The method involves linearization of nonlinear dynamic equations about an operating point on a speed line, and then application of the first method of Lyapunov to determine the stability of the non-linear system from the stability of the linear system. The method is applied to a simulation of the J85 compressor, which utilizes stage stacking and lumped volume techniques for the interstage regions to simulate steady-state and dynamic compressor performance. The stability boundary predicted by the digital simulation compares quite well with the stall line predicted by a dynamic simulation of the J85 compressor programmed on the analog computer. Since previous studies have shown that the analog-predicted stall line agrees well with the stall line of the compressor, the digital method presented is also a good means of estimating the stall line.

PREDICTION OF COMPRESSOR STALL FOR DISTORTED AND UNDISTORTED FLOW BY USE OF A MULTISTAGE COMPRESSOR SIMULATION ON THE DIGITAL COMPUTER
Daniele, C. J.; Teren, F.
National Aeronautics and Space Administration. Lewis Research Center. Cleveland, Ohio.
1974, 12p.
NASA-TM-X-71633; E-8171
N75-13190
A simulation technique is presented for the prediction of compressor stall for axial-flow compressors for clean and distorted inlet flow. The simulation is implemented on the digital computer and uses stage stacking and lumped-volume gas dynamics. The resulting nonlinear differential equations are linearized about a steady-state operating point, and a Routh-Hurwitz stability test is performed on the linear system matrix. Parallel compressor theory is utilized to extend the technique to the distorted inlet flow problem. The method is applied to the eight-stage J85-13 compressor.

A COMPUTER PROGRAM FOR THE ANALYSIS AND DESIGN OF TURBOMACHINERY
Davis, W. R.
Carleton Univ., Ottawa (Ontario). Div. of Aerothermodynamics.
Sep 1971, 119p. refs.
ME/A-71-5
N71-37385
A streamline curvature method for analyzing the flow conditions and performance of a turbomachine at design and off-design conditions is described. The inviscid rotational axi-symmetric flow field, with enthalpy and entropy gradients permitted, is determined by an iterative solution. A computer program for the above is described in detail and a FORTRAN listing included.

A MATRIX METHOD APPLIED TO THE ANALYSIS OF THE FLOW IN TURBOMACHINERY
Davis, W. R.
Carleton Univ., Ottawa (Ontario) Div. of Aerothermodynamics.
Sep 1971, 111p. refs.
ME/A-71-6
N72-12217
A matrix technique for analyzing the flow conditions and performance of a turbomachine is described. The inviscid, rotational compressible flow field, with enthalpy and entropy gradients permitted, is determined by an iterative solution. A computer program for the above is described in detail and a FORTRAN listing provided.

AXIAL FLOW COMPRESSOR ANALYSIS USING A MATRIX METHOD
Davis, W. R., Millar, D. A.
Carleton Univ., Ottawa (Ontario)
Feb 73, 92p.
ME/A-73-1
N73-22723

COMPARISON OF THE MATRIX AND STREAMLINE CURVATURE METHODS OF AXIAL FLOW TURBOMACHINERY ANALYSIS, FROM A USER'S POINT OF VIEW.
Davis, W. R.; Millar, D. A. J.
Carrier Corp., Syracuse, NY
ASME Pap. N 74-WA/GT-4 for Meet. Nov 17-22 1974, p.10

TRANSONIC FLOW ANALYSIS IN AXIAL-FLOW TURBOMACHINERY CASCADES BY A TIME-DEPENDENT METHOD OF CHARACTERISTICS.
Delaney, R. A.; Kavanagh, P.
ASME Pap. N 75-GT-8 For Meet. Mar 2-6 1975, p.8

FORTRAN PROGRAM TO GENERATE ENGINE INLET FLOW CONTOUR MAPS AND DISTORTION PARAMETERS
Dicus, J. H.
National Aeronautics and Space Administration. Lewis Research Center, Cleveland, Ohio.
Feb 1974, 63p. refs.
NASA-TM-X-2967; E-7572
N74-15464
A computer program is presented and described that generates jet engine inlet flow contour maps and inlet flow distortion parameters. The program input consists of an array of measurements describing the flow conditions at the engine inlet. User-defined distortion parameters may be calculated.

DESIGN AND TESTING OF THE FLOW-THROUGH PART OF A GAS-TURBINE ENGINE
Dondoshanskii, V. K.; Dergach, V. F.
Foreign Technology Div. Wright-Patterson AFB, Ohio.
20 Jan 74, 21p.
FTD-HC-23-1627-74
AD-782 542

SIMULATION METHOD APPLIED TO RELIABILITY ANALYSIS OF TURBOJET ENGINE COMPONENTS OPERATING IN AIRLINE SERVICE
Dorsey, R. S.; Truscott, H. A.
General Electric Co., Cincinnati, Ohio.
Annals of Assurance Sciences, V. 8, July 7-9 1969, p.524-9 8th Reliability and Maintainability Conference.

ADVANCED SMALL AXIAL TURBINE TECHNOLOGY PROGRAM
Due, H. F.; Rogo, C.
Teledyne Cae Toledo Ohio
Feb 74, 167p.
TCAE-1329
AD-780 152
The report presents the Phase II experimental investigations of the Phase I turbine design aerodynamics accomplished using a non-rotating, annular sector cascade. Phase II is the second phase of a five-phase thirty-one month program. The overall program is arranged to isolate the variables which affect turbine flow, and to develop both empirical data and design judgment criteria which can be incorporated into isolate the variables which affect turbine flow, and to develop both empirical data and design judgment criteria which can be incorporated into the design technique to improve the accuracy of predicting losses, flow conditions, velocity triangles and design point matching.

IMPROVEMENTS TO THE AINLEY-MATHIESON METHOD OF TURBINE PERFORMANCE PREDICTION
Dunham, J.; Came, P. M.
National Gas Turbine Estab., Hantshire, England
J Eng Power, Trans ASME V 92 Ser A N 3 July 1970, p.252-6

EFFECTS OF HEAT SOAKAGE IN AXIAL FLOW COMPRESSORS.
Elder, R.
Cranfield Inst. of Technol, Bedford, England
ASME Pap. N 74-WA/GT-5 for Meet. Nov 17-22 1974, p.8

METHOD FOR THE NUMERICAL CALCULATION OF VELOCITY DISTRIBUTION FOR A BLADE CASCADE ROTATING IN A PERFECT INCOMPRESSIBLE FLUID.
Eremeef, L. R.
Period Polytech, Mech Eng V 18 N 1 1974, p.53-59

UNSTABLE CONDITIONS OF TURBODYNAMICS. ROTATING STALL
Ershov, V. N.
Foreign Technology Div. Wright-Patterson AFB, Ohio.
12 Aug 71, 254p.
FTD-MT-24-04-71
AD-731 355
The book gives the results of experimental and theoretical research of unstable conditions in turbodynamos. In this case primary attention is paid to conditions of rotating stall. The characteristics of the procedure of experimental research and of the low-inertia apparatus used in this case are described. Some methods of expansion of the area of stable conditions are given. This book is intended for scientific workers, graduate students, and engineer-designers working in the area of research and design of gas-turbine engines, compressors, fans, and pumps.

SOME OBSERVED EFFECTS OF PART-SPAN DAMPERS ON ROTATING BLADE ROW PERFORMANCE NEAR DESIGN POINT
Esgar, G. M.; Sandercock, D. M.
National Aeronautics and Space Administration. Lewis Research Center, Cleveland, Ohio.
Jan 1973, 27p. refs.
NASA-TM-X-2696; E-7067
N73-14985
Detailed measured radial distributions of flow parameters for eight rotors with part-span dampers are used to study the effects of dampers on rotor performance and flow parameters at near design operation. All rotors had a blade tip diameter of about 20 in. and operated at a blade tip speed of about 1400 ft/sec. Several examples demonstrate that, when the local loss variations in the damper flow region are included in an aerodynamic design or analysis procedure, the computed spanwise distributions of flow parameters compare closely with measured distributions.

EFFECTS OF FREE STREAM TURBULENCE ON BLADE PERFORMANCE IN A COMPRESSOR CASCADE
Evans, B.J.
Cambridge Univ. (England). Dept. of Engineering.
1971, 48p. refs.
CUED/A-Turbo/TR-26
N72-13266
Boundary layer behavior on the stator blades of an axial flow compressor were studied under the effects of free stream turbulence generated by the passage of upstream rotor blades across the flow field, at low Reynolds' numbers. These conditions were simulated in a specially designed two-dimensional linear cascade. A detailed study of the boundary layer behavior was made up to turbulence levels of 4% using flow visualization and hot wire anemometry techniques. Reliable data on separation bubble length and on natural attached transition length and position is presented in a directly applicable form, and the necessity of considering the turbulence structure is demonstrated by an experimental investigation into the isolated effects of scale upon transition.

HIGHLY LOADED MULTI-STAGE FAN DRIVE TURBINE PLAIN BLADE CONFIGURATION DESIGN
Evans, D. C.; Wolfmeyer, G. W.
General Electric Co., Cincinnati, Ohio.
Feb 72, 112p.
NASA-CR-1964; GE-R71-AEG-242
N72-17845
The constant-inside-diameter flowpath was scaled for testing in an existing turbine test facility. Blading detailed design is discussed, and design data are summarized. Predicted performance maps are presented. Steady-state stresses and vibratory behavior are discussed and the results of the mechanical design analysis are presented.

EXPERIMENTAL INVESTIGATION OF A 4 1/2-STAGE TURBINE WITH VERY HIGH STAGE LOADING FACTOR. 1 TURBINE DESIGN.
Evans, D. C.; Hill, J. M.
General Electric Co., Cincinnati, Ohio.
Jan 73, 126p.
NASA-CR-2140
N73-16770
The results of the Task 1 and 2 turbine design work are reported. Preliminary design is discussed. Blading detailed design data are summarized. Predicted performance maps are presented. Steady-state stresses and vibratory behavior are discussed, and the results of the mechanical design analysis are presented.

METHOD FOR DETERMINING COMPONENT MATCHING AND OPERATING CHARACTERISTICS FOR TURBOJET ENGINES.
Evans, D. G.
National Aeronautics and Space Administration. Lewis Research Center. Cleveland, Ohio
Jan 70, 23p.
NASA-TM-X-1945; E-5316
N70-17957

COMPUTER PROGRAM DESCRIBING TURBINE AERODYNAMIC REQUIREMENTS , APPROXIMATE EXTERNAL BLADE GEOMETRIES, AND COOLANT FLOW REQUIREMENTS FOR A TWO STAGE AXIAL FLOW TURBINE.
Evans, D. G.; Furgalus, K. A.; Vanco, M. R.
National Aeronautics and Space Administration. Lewis Research Center. Cleveland, Ohio.
Apr 71, 59p.
NASA-TM-X-2229; E-6014
N71-22568

CALCULATION OF THE DEVELOPMENT OF TURBULENT BOUNDARY LAYERS WITH A TURBULENT FREESTREAM
Evans, R. L.; Horlock, J. H.
Cambridge Univ, Engl.
J Fluid Eng Trans ASME V 96, Ser 1 N 4, Dec 1974, p.348-352

VIBRATION CHARACTERISTICS OF BLADED DISC ASSEMBLIES
Ewins, D. J.
Imperial Coll of Science and Technol, London, Engl
J Mech Eng Sci V 15 N 3, Jun 1973, p.165-186

EFFECT OF OUTER CASING TREATMENT AND TIP CLEARANCE ON STALL MARGIN OF A SUPERSONIC ROTATING CASCADE.
Fabri, J.; Reboux, J.
ASME Pap. N 75-GT-95 for Meet. Mar 2-6 1975, p.7

DIGITAL COMPUTER METHODS FOR PREDICTION OF GAS TURBINE DYNAMIC RESPONSE
Fawke, A. J.; Saravanamuttoo, H. I. H.
SAE Pap. 710550 for Meet. June 7-11 1971, p.9

EXPERIMENTAL INVESTIGATION OF METHODS FOR IMPROVING THE DYNAMIC RESPONSE OF A TWIN-SPOOL TURBOJET ENGINE
Fawke, A. J.; Saravanamuttoo, H. I. H.
Engineering Res Station, Gas Council, Newcastle, England
J Eng Power, Trans ASME V 93, Ser A N 4, Oct 1971, p.418-24

EXPERIMENTAL VERIFICATION OF A DIGITAL COMPUTER SIMULATION METHOD FOR PREDICTING GAS TURBINE DYNAMIC BEHAVIOUR
Fawke, A. J.; Saravanamuttoo, H. I. H.; Holmes, M.
Gas Council Engineering Res Station, Newcastle Upon Tyne, Eng.
Inst. Mech. Eng. (London), Proc. V 186 Pap. N 27, 1972, p. 323-329

DIGITAL COMPUTER SIMULATION OF THE DYNAMIC RESPONSE OF A TWIN SPOOL TURBOFAN WITH MIXED EXHAUSTS
Fawke, A. J.; Saravanamuttoo, H. I. H.
Br Gas Corp, Newcastle, Engl
AERON J V 77, N 753, Sep 1973, p.471-478

GENENG 2: A PROGRAM FOR CALCULATING DESIGN AND OFF-DESIGN PERFORMANCE OF TWO- AND THREE-SPOOL TURBOFANS WITH AS MANY AS THREE NOZZLES.
Fishbach, L. H.; Koenig, R. W.
National Aeronautics and Space Administration. Lewis Research Center, Cleveland, Ohio.
Feb 1972, 187p. refs.
NASA-TN-D-6553; E-6356
N72-18784
A computer program which calculates steady-state design and off-design jet engine performance for two- or three-spool turbofans with one, two, or three nozzles is described. Included in the report are complete FORTRAN 4 listings of the program with sample results for nine basic turbofan engines that can be calculated: (1) three-spool, three-stream engine; (2) two-spool, three-stream, boosted-fan engine; (3) two-spool, three-stream, supercharged-compressor engine; (4) three-spool, two-stream engine; (5) two-spool, two-stream engine; (6) three-spool, three-stream, aft-fan engine; (7) two-spool, three-stream, aft-fan engine; (8) two-spool, two-stream, aft-engine; and (9) three-spool, two-stream, aft-fan engine. The simulation of other engines by using logical variables built into the program is also described.

A COMPUTER PROGRAM FOR THE SPECIFICATION OF AXIAL COMPRESSOR AIRFOILS
Frost, G. R.; Hearsey, R. M.; Wennerstrom, A. J.
Aerospace Research Labs Wright-Patterson AFB, Ohio
Dec 72, 168p.
ARL-72-0171
AD-756 879
The report describes the analysis in, and the use of, a computer program which has been developed for use in the design of axial compressor airfoils suitable for operation at high subsonic and supersonic Mach numbers. Four rather versatile camber line shapes and two thickness distributions are mathematically derived. These camber lines provide the capability of defining a wide variety of blades, from those of continuously positive camber to the so-called 'S-blades', including many of the intermediate possibilities. A method is presented whereby the airfoils are specified on arbitrary axisymmetric streamsurfaces and then accurately redetermined in Cartesian coordinates on planes normal to the stacking axis.

FORTRAN PROGRAM FOR PREDICTING OFF-DESIGN PERFORMANCE OF CENTRIFUGAL COMPRESSORS
Galvas, M. R.
National Aeronautics and Space Administration. Lewis Research Center, Cleveland, Ohio.
Nov 1973, 59p. refs.
NASA-TN-D-7487; E-7480
N74-10024
A FORTRAN program for calculating the off-design performance of centrifugal compressors with channel diffusers is presented. Use of the program requires complete knowledge of the overall impeller and diffuser geometries. Individual losses are computed using analytical equations and empirical correlations which relate loss levels to velocity diagram characteristics and overall geometry. On a given speed line compressor performance is calculated for a range of inlet velocity levels. At flow rates between surge and choke, individual efficiency decrements, compressor overall efficiency, and compressor total pressure ratio are tabulated. An example case of performance comparison with a compressor built by a commercial engine manufacturer is presented to demonstrate the correlation with limited experimental data.

DYNAMIC SIMULATION. TOOL FOR ENGINE COMPONENT DESIGN
Gardner, W. B.; Sampl, F.
SAE-Paper 690386 for Meet. Apr 21-24 1969, p.7

THE STUDY OF A CIRCUMFERENTIALLY NONUNIFORM FLOW IN FRONT OF AN AXIAL-FLOW COMPRESSOR STAGE
Ginzburg, S. I.; Suslennikov, L. A.
Foreign Technology Div. Wright-Patterson AFB, Ohio
23 Mar 73, 30p.
FTD-MT-24-1787-72
AD-759 247
The question of the operation of an axial-flow compressor in a circumferentially nonuniform flow is of considerable practical interest both with respect to the effect of nonuniformity on the gas-dynamic parameters of the stages and their stability and with respect to the determination of the variable forces which arise on the impeller blades.

COMPUTER PROGRAM FOR PRELIMINARY DESIGN ANALYSIS OF AXIAL-FLOW TURBINES
Glassman, A. J.
National Aeronautics and Space Administration. Lewis Research Center. Cleveland, Ohio.
Mar 72, 38p.
NASA-TN-D-6702; E-6603
N72-18774
The program method is based on a mean-diameter flow analysis. Input design requirements include power or pressure ratio, flow, temperature, pressure, and speed. Turbine designs are generated for any specified number of stages and for any of three types of velocity diagrams (symmetrical, zero exit swirl, or impulse), exit turning vanes can be included in the design. Program output includes inlet and exit annulus dimensions, exit temperature and pressure, total and static efficiencies, blading angles, and last-stage critical velocity ratios. The report presents the analysis method, a description of input and output with sample cases, and the program listing.

NEW TECHNOLOGY IN TURBINE AERODYNAMICS.
Glassman, A. J.; Moffitt, T. P.
National Aeronautics and Space Administration. Lewis Research Center. Cleveland, Ohio.
1972, 34p.
NASA-TM-X-68115; E-7059
N72-28795
A cursory review is presented of some of the recent work that has been done in turbine aerodynamic research at Nasa-Lewis Research Center. Topics discussed include the aerodynamic effect of turbine coolant, high work-factor (ratio of stage work to square of blade speed) turbines, and computer methods for turbine design and performance prediction. An extensive bibliography is included. Experimental cooled-turbine aerodynamics programs using two-dimensional cascades, full annular cascades, and cold rotating turbine stage tests are discussed with some typical results presented. Analytically predicted results for cooled blade performance are compared to experimental results. The problems and some of the current programs associated with the use of very high work factors for fan-drive turbines of high-bypass-ratio engines are discussed. Turbines currently being investigated make use of advanced blading concepts designed to maintain high efficiency under conditions of high aerodynamic loading. Computer programs have been developed for turbine design-point performance, off-design performance, supersonic blade profile design, and the calculation of channel velocities for subsonic and transonic flow fields. The use of those programs for the design and analysis of axial and radial turbines is discussed.

AN APPLICATION OF STREAMLINE CURVATURE METHODS TO THE CALCULATION OF FLOW IN A MULTISTAGE AXIAL COMPRESSOR
Glenny, D. E.
Aeronautical Research Labs., Melbourne (Australia).
May 74, 50p.
ARL/ME-NOTE-346
N75-16558
The performance of a multistage axial compressor, predicted by a computer model using a streamline curvature technique, is considered in terms of the flow development within the compressor and the assumptions inherent in the program. An analysis of the results shows that provided adquate allowance is made for the effects of the growth of the annular boundary layers and the development of stall regions within the blade rows, the overall predicted stage by stage flow characteristics are consistent with those known to exist in the actual compressor. However, at present, it is not possible to determine the necessary annular blockage through the machine without resort to actual experimental data in the flow model. In addition, because of the absence of a suitable means for allowing for secondary flow in the model, it is not possible to consider the development of the flow profiles in the region of the annulus walls.

OFF-DESIGN BEHAVIOR FOR AXIAL FLOW COMPRESSOR STAGES WITH INVARIABLE AND VARIABLE GEOMETRY BLADES
Grahl, K.; Tabakoff, W.
Cincinnati Univ Ohio Dept of Aerospace Engineering
Sep 73, 67p.
73-39
AD-767 265
An application of a computer program is described for studying the off-design performance for single and multistage axial flow compressors. The calculation method allows one to determine the stage characteristics as a function of revolutions. This can be accomplished in a very short time with only a few geometrical and aerodynamical input data. The flow is described by the well-known streamline curvature method. The necessity of a good loss model is shown. Limitations of this method are described. Some computation examples for the first and fifth stages of a ten-stage Nasa axial flow compressor are presented. The stage and overall characteristics for different compressors are considered for invariable and variable blade geometry. The off-design analytical results are compared with existing experimental results.

SURFACE VORTICITY ANALYSIS OF THREE-DIMENSIONAL FLOW THROUGH STRONGLY SWEPT TURBINE CASCADES.
Graham, D.; Lewis, R. I.
Univ of Newcastle Upon Tyne, Engl
J Mech Eng Sci V 16, N 6, Dec 1974, p.425-433

SURGE AND ROTATING STALL IN AXIAL FLOW COMPRESSORS - 1, 2.
Greitzer, E. M.
ASME Pap. N 75-GT-9 for Meet. Mar 2-6 1975, 9p. N 75-GT-10, p.13

PERFORMANCE PREDICTION OF TRANSONIC BLADINGS WITH HIGH DEFLECTION & LOW ASPECT RATIO.
Griepentrog, H.
Von Karman Inst. for Fluid Dynamics. Rhode Saint-genese (Belgium)
Dec 69, 243p.
VKI-TN-59 N70-34929

PERFORMANCE PREDICTION FOR HIGH TURNING LOW ASPECT RATIO STATOR CASCADES IN THE TRANSONIC REGIME
Griepentrog, H. F. L.
Karman Inst, Genese, Belgium
J Eng Power, Trans ASME V92, Ser A N 4, Oct 1970, p.390-8

DESIGNING JET AIRCRAFT WIND- TUNNEL TEST PROGRAMS WITH PROPULSION SYSTEM SIMULATION
Grunnet, J. L.
Fluidyne Engineering Corp, Minneapolis, Minn
J Aircraft V 8, N 6, June 1971, p.421-7

APPLICATION OF THE MULTIPLE-SCALE CONCEPT TO THE FLOW IN AN AXIAL FLOW TURBOMACHINE
Guiraud, J. P.; Zeytounian, R. K. H.
Off Natl D'Etudes et de Rech Aerosp, Chatillon-Sous-Bagneux
Int J Eng Sci V 12, N 4, Apr 1974, p.311-330

AN EXPERIMENTAL INVESTIGATION OF THE EFFECTS OF MASSIVE FILM COOLING ON THE AERODYNAMICS OF A TURBINE AIRFOIL.
Hartsel, J. E.
Ohio State Univ Research Foundation Columbus
Aug 70, 176p.
AFAPL-TR-70-97
AD-745 575
The effects of massive film cooling through multiple rows of discrete holes on the aerodynamics of a typical two-dimensional turbine airfoil have been studied experimentally using a single blade positioned in a contoured channel. The channel walls, shaped to simulate the presence of adjacent airfoils in a cascade, were both porous and movable to allow adjustment of the flow direction and airfoil surface pressure. Electrically heated air was used as the primary flow while room temperature coolant air was emitted from five rows of discrete holes on each airfoil surface to film cool the regions aft of mid-chord. Hole geometries angled in the spanwise, chordwise, and vertical directions were employed to achieve coolant-to-inlet mass rate ratios as high as 0.05 for blowing from the single blade.

HEAT TRANSFER FROM A SHROUDED DISK SYSTEM WITH A RADIAL OUTFLOW OF COOLANT.
Haynes, C. M.; Owen, J. M.
Univ of Sussex, Engl.
J Eng Power Trans ASME V 97, Ser A N 1, Jan 1975, p.28-36

A COMPUTER PROGRAM FOR AXIAL COMPRESSOR DESIGN. VOLUME I. THEORY DESCRIPTIONS, AND USERS INSTRUCTIONS.
Hearsey, R. M.
Dayton Univ Ohio Research Inst
Jul 73, 97p.
AFAPL-TR-73-66-Vol-1
AD-764 733
A computer program for the design of axial compressors is presented. It comprises of three principal sections: two alternative means of determining blade geometry, and an aerodynamic computation for the flow through the compressor. One method of determining blade geometry revolves around the use of various analytic meanlines for the blade sections, and leads to the aerodynamic analysis of the flow through specified blading. The other method consists of creating arbitrary blade sections to follow the flow directions previously determined in an aerodynamic design calculation. The aerodynamic design section incorporates a loss calculation routine that may be used to estimate the design point performance of the compressor. The report describes the computer program, and gives all information necessary to use it.

THEORETICAL FUNDAMENTALS OF CALCULATING STATIONARY PERFORMANCE CHARACTERISTICS OF TURBOJET ENGINES.
Heise, G.
Dornier-Werke G.M.B.H., Friedrichshafen (West Germany)
Apr 69, 44p.
Rept-69/2
N69-38088

THE UNSTEADY RESPONSE OF AN AXIAL FLOW TURBOMACHINE TO AN UPSTREAM DISTURBANCE.
Henderson, R. E.
Pennsylvania State Univ. University Park Ordnance Research Lab.
27 Oct 72, 177p.
TM-72-218
AD-759 029
The results of a study to determine the unsteady response of an axial flow turbomachine rotor to spatial variations in its inlet flow are presented. This study includes the development of a theoretical analysis which permits the unsteady lift of a blade row to be expressed in terms of the usual blade design parameters. With this solution, which includes the unsteady contribution of the entire blade row, the blade spacing, stagger angle and camber can be selected to give a desired level of unsteady response. Comparisons of this solution with existing solutions for blade rows with specified geometries are presented.

APPROXIMATE ANALYSIS OF THE UNSTEADY LIFT ON AIRFOILS IN CASCADE.
Henderson, R. E.; Horlock, J. H.
Pa State Univ., State College
J Eng Power, Trans ASME V 94, Ser A N 4, Oct 1972, p.233-240

EFFECT OF BLADE ROW GEOMETRY ON AXIAL FLOW ROTOR UNSTEADY RESPONSE TO INFLOW DISTORTIONS.
Henderson, R. E.; Bruce, E. P.
Pennsylvania State Univ. State College Applied Research Lab.
18 Jul 74, 34p.
TM-74-224
AD-785 276
Employing a recently developed experimental apparatus, the authors have measured the unsteady response of an isolated axial flow fan rotor to inflow distortions. The measured quantities include the unsteady normal force and pitching moment on a segment of a single blade operating at various mean angles of attack and the associated distribution of time-mean total pressure change across the rotor. These results are part of a larger program that includes variations in the mean angle of attack in reduced frequency omega, in blade stagger angle, and in blade row space-to-chord ratio (S/C). The effect of variations in omega and S/C on the unsteady response are presented.

VIBRATION ANALYSIS OF ROTATING COMPRESSOR BLADES.
Henry, R.; Lalanne, M.
Inst. Natl Des Sci Appl. Villeurbanne, Fr.
J Eng Ind., Trans ASME V 9, Ser B N 3, Aug 1974, p.1028-1035

END-WALL BOUNDARY LAYERS IN AXIAL COMPRESSORS.
Hirsch, C. H.
Vrije Univ. Brussels, Belg.
ASME Pap. N 74-GT-72 for Meet. Mar 31-Apr 4 1974, p.14

A FINITE ELEMENT METHOD FOR FLOW CALCULATIONS IN TURBOMACHINES
Hirsch, C.; Warzee, G.
Brussels Univ. (Belgium). Dept. of Fluid Mechanics.
Jul 74, 76p.
VUB-STR-5
N75-10946
The pitch-averaged equations in a meridional plane of a turbomachine are derived from the general flow equations. Assuming axisymmetry, these equations are solved, in the case of axial compressors, with the finite element method. The finite element method and the calculation procedure are described. The method is limited to subsonic meridional velocities allowing the treatment of transonic stages. Comparisons with experimental results for a transonic single compressor stage and a subsonic 6-stage axial compressor show excellent agreement.

FLOW MODELS FOR TURBOMACHINES.
Horlock, J. H.; Marsh, H.
Cambridge Univ. (England). Dept. of Engineering.
1971, 36p. refs
CUED/A-Turbo/TR-25
N72-13265
The equations for the flow through cascades of blades are averaged across the pitch and then compared with those for the flow through hypothetical models of blade rows - a closely spaced blade row, an axisymmetric flow with body forces and the flow on a mean stream surface. It is shown that these models can provide an accurate representation for the overall flow changes across a blade row, but they cannot give an exact local matching with the averaged properties of the flow in the real cascade. The nature of the blade forces in the real flow and the body forces in the hypothetical flow is discussed.

ANNULUS WALL BOUNDARY LAYERS IN TURBOMACHINES.
Horlock, J. H.; Perkins, H. J.
Advisory Group for Aerospace Research and Development Paris (France).
May 74, 69p.
AGARD-AG-185
AD-780 787
The paper reviews a substantial body of fundamental and applied research work on the subject of annulus wall boundary layers in turbomachines. A product of this work is a method for predicting, with reasonable accuracy, the full three-dimensional boundary layer that develops through a single blade row. The limitations of the method are discussed in relation to the experimental information available to the authors. In Part 2 the fundamental work is subjected to scrutiny in order to extract a boundary layer calculation method suitable for estimating the effects of annulus wall boundary layers on the performance of turbomachines. It is shown that a great many assumptions and simplifications are necessary at the present time but that reasonably satisfactory estimates are nevertheless possible. A computer program for the estimation of annulus blockage is listed.

PROPAGATING STALL IN COMPRESSORS WITH POROUS WALLS.
Horlock, J. H.; Lakhwani, C. M.
ASME Pap. N 75-GT-59 for Meet. Mar 2-6 1975, p.8

CONTRIBUTION TO THE QUASI-ROTATIONAL SYMMETRICAL CALCULATION OF THREE DIMENSIONAL FLOW IN AXIAL BLADINGS CONSIDERING BLOCKING PH.D. THESIS (BEITRAG ZUR QUASIROTATIONSSYMMETRISCHEN BERECHNUNG DER DREIDIMENSIONALEN STROEMUNG IN AXIALEN BESCHAUFELUNGEN UNTER BERUECKSICHTIGUNG DER VERSPERRUNG)
Hourmouziadis, J.
Technische Univ., Berlin (West Germany)
1971, 98p. refs.
N72-15261
A general dimensionless form of the radial pressure equation is developed and used in an approximative theorem for blade pressure to calculate the cascade flow. Comparison of calculated and measured flow values shows that: A generalized radial pressure equation represents the actual flow dynamics inside the axial fold only insufficiently because radial velocity is neglected; blade force exerts a strong effect on the radial pressure gradient inside the cascade; variable radial pressure gradient and profile form determine axial velocity distribution; and cascade produced perturbations diminish quickly in front and after the cascade. It is concluded that blade positioning is of utmost importance on cascade flow.

FEARED COMPRESSOR STALL OF JET TURBINE ENGINES.
Hufnagel, S.
Wehrtechnik N 4, Apr 1969, p.138-141

REQUIREMENTS FOR DIGITAL COMPUTER SIMULATION OF GAS TURBINE PROPULSTION SYSTEM PERFORMANCE. PHASE I. STEADY-STATE AND TRANSIENT ENGINE PERFORMANCE SIMULATION.
Hutcheson, L.; Armstrong, W. C.; Cooper, C. B.
Arnold Engineering Development Center Arnold Air Force Station Tenn
Mar 71, 34p.
AEDC-TR-71-24
AD-720 803
Present and near-future requirements for the addition of digital computer simulation of gas turbine engine steady-state and transient performance to the present Engine Test Facility and Propulsion Wind Tunnel Facility digital data capability were determined based on information and guidance provided by the Air Force Aero Propulsion Laboratory and various gas turbine engine manufacturers. During Phase I of this study,

digital computer high-speed core memory size and throughput times were determined and are presented for several modern steady-state and transient mathematical model simulation programs. Display requirements were also determined and are presented for full utilization of the mathematical model results, off-line and on-line. Some preliminary results on dynamic compressor mathematical models are discussed.

A TWO-DIMENSIONAL CASCADE TEST OF AN AIR-COOLED TURBINE NOZZLE. PART 2 - ON THE TEMPERATURE DISTRIBUTIONS OF A CONVECTION-COOLED BLADE AS DETERMINED BY NUMERICAL CALCULATION AND BY ANALOGUE SIMULATION TEST.
Inoue, S.; Mimura. F.; Nouse, H.; Takahara, K.; Yoshida, T.
National Aerospace Lab., Tokyo (Japan).
Jan 71, 24p.
NAL-TR-232
N71-33549

SIMULATION STUDY OF TRANSIENT PERFORMANCE MATCHING OF TURBOFAN ENGINE USING AN ANALOGUE COMPUTER TO EVALUATE ITS USEFULNESS AS DESIGN TOOL.
Itoh, M.; Ishigaki, T.; Sagiya, Y.
Ishikawajima-Harima Heavy Ind. Co., Tokyo, Jap.
ASME Pap. N 74-GT-50 for Meet. Mar 31-Apr 4 1974, p.6

CONVERSION OF INLET TEMPERATURE DISTORTIONS TO VORTICITY FOR AN AXIAL-FLOW COMPRESSOR.
Iverson, M. M.
Naval Postgraduate School Monterey Calif.
Jun 72, 92p.
AD-745 852
A survey of the literature on pressure, temperature and foreign gas inlet distortions is made. For interpretation of the influence of inlet distortion on engine stability, a break is made with distortion methods which use temperature or pressure distortion maps. The distortion in the form of temperature contours is transformed to vorticity using appropriate equations derived for both incompressible and compressible flow. A numerical solution of Poisson's equation to describe secondary flow arising from convected vorticity is presented.

COMPRESSOR SENSITIVITY TO TRANSIENT AND DISTORTED TRANSIENT FLOWS. VOLUME II. MATHEMATICAL DETAILS AND COMPUTER PROGRAMS.
Jansen, W.; Swarden, M. C.; Carlson, A. W.
Northern Research and Engineering Corp. Cambridge Mass.
Jan 71, 184p.
NREC-1149-2
AD-728 024
The report is the second volume of a report describing the results of an analytical investigation of compressor sensitivity, conducted under Contract No. N00019-69-C-0602 for the Department of the Navy. Naval Air Systems Command. The objectives of the investigation were to develop an analytical model which simulates compressor response to spatial and temporal flow disturbances; to establish by application of the model the importance of various flow mechanisms, and their relationship to design variables, in determining compressor response; and to establish some design trends by means of a limited parametric study. This volume contains the detailed mathematical development of the models and the theoretical background of the equations that were utilized; the results of the studies are described in general terms in Volume 1.

SIMULATION OF TURBINE STAGE OPERATION WITH WORKING MEDIA DIFFERING FROM ACTUAL ONES.
Kalinin, G. E.
Energomashinostroenie N 8 Aug 1972, p.25-27

THE THEORY OF TURBO-MACHINES (SELECTED CHAPTERS)
Kirillov, I. I.
Foreign Technology Div. Wright-Patterson AFB Ohio
24 Jun 74, 330p.
FTD-MT-24-423-74
AD-784 180

EFFECT OF INDUCER INLET AND DIFFUSER THROAT AREAS ON PERFORMANCE OF A LOW PRESSURE RATIO SWEPTBACK CENTRIFUGAL COMPRESSOR.
Klassen, H. A.
National Aeronautics and Space Administration. Lewis Research Center. Cleveland, Ohio
Jan 75, 26p.
NASA-TM-X-3148; E-8010
N75-13869
A low-pressure-ratio centrifugal compressor was tested with nine combinations of three diffuser throat areas and three impeller inducer inlet areas which were 75, 100, and 125 percent of design values. For a given inducer inlet area, increases in diffuser area within the range investigated resulted in increased mass flow and higher peak efficiency. Changes in both diffuser and inducer areas indicated that efficiencies within one point of the maximum efficiency were obtained over a compressor specific speed range of 27 percent. The performance was analyzed of an assumed two-spool open-cycle engine using the 75 percent area inducer with a variable area diffuser.

GENENG: A PROGRAM FOR CALCULATING DESIGN AND OFF-DESIGN PERFORMANCE FOR TURBOJET AND TURBOFAN ENGINES.
Koenig, R. W.; Fishbach, L. H.
National Aeronautics and Space Administration. Lewis Research Center, Cleveland, Ohio.
1972, 161p.
NASA-TN-D-6552; E-5867
N72-18783
A computer program entitled GENENG employs component performance maps to perform analytical, steady state, engine cycle calculations. Through a scaling procedure, each of the component maps can be used to represent

a family of maps (different design values of pressure ratios, efficiency, weight flow, etc.) Either convergent or convergent-divergent nozzles may be used. Included is a complete FORTRAN 4 listing of the program. Sample results and input explanations are shown for one-spool and two-spool turbojets and two-spool separate- and mixed-flow turbofans operating at design and off-design conditions.

OVERALL AND BLADE-ELEMENT PERFORMANCE OF A TRANSONIC COMPRESSOR STAGE WITH MULTIPLE-CIRCULAR-ARC BLADES AT TIP SPEED OF 419 METERS PER SECOND.
Kovich, G.; Reid, L.
National Aeronautics and Space Administration. Lewis Research Center, Cleveland, Ohio.
Mar 73, 111p.
NASA-TM-X-2731; E-6823
N73-19995

PERFORMANCE OF TRANSONIC FAN STAGE WITH WEIGHT FLOW PER UNIT ANNULUS AREA OF 198 KILOGRAMS PER SECOND PER SQUARE METER (40.6(LB/SEC)/SQ FT)
Kovich, G.; Moore, R. D.; Urasek, D. C.
National Aeronautics and Space Administration. Lewis Research Center. Cleveland, Ohio.
Nov 1973, 92p. refs.
NASA-TM-X-2905; E-7205
N74-10027
The overall and blade-element performance are presented for an air compressor stage designed to study the effect of weight flow per unit annulus area on efficiency and flow range. At the design speed of 424.8 m/sec the peak efficiency of 0.81 occurred at the design weight flow and a total pressure ratio of 1.56. Design pressure ratio and weight flow were 1.57 and 29.5 kg/sec (65.0 lb/sec), respectively. Stall margin at design speed was 19 percent based on the weight flow and pressure ratio at peak efficiency and at stall.

CALCULATION OF THE DISTRIBUTION OF LOSSES OVER THE SPAN OF AN AXIAL-FLOW COMPRESSOR BLADE
Beknev, V. S.; Kozhevnikov, V. Yu.
Foreign Technology Div. Wright-Patterson AFB, Ohio
2 Aug 72, 13p.
FTD-MT-24-245-72
AD-750 931
On the basis of experimental data, relationships are obtained which make it possible to calculate the losses at each cross section with respect to the height of an axial compressor blade. The obtained results permit a judgment to be made concerning the influence of various cascade parameters upon the distribution of losses with respect to blade height.

VISUALIZATION STUDY OF FLOW IN AXIAL FLOW INDUCER.
Lakshminarayana, B.
Pa. State Univ., University Park.
J Basic Eng., Trans ASME V 94, Ser D N 4, Dec 1972, p.777-787

METHOD OF MEASURING THREE-DIMENSIONAL ROTATING WAKES BEHIND TURBOMACHINERY ROTORS.
Lakshiminarayana, B.; Poncet, A.
Pa. State Univ., University Park.
J Fluid Eng Trans ASME V 96, Ser 1 N 2, Jun 1974, p.87-91

FLUID MECHANICS, ACOUSTICS, AND DESIGN OF TURBOMACHINERY, PART 1.
Lakshminarayana, ed., B.; Britsch, W. R.; Gearhart, W. S.
Pennsylvania State Univ., University Park.
1974, 419p. refs.
NASA-SP-304-Pt-1; LC-72-600087-Pt-1
N75-11174
A conference was conducted to analyze the effects of air flow on turbomachinery design. The subjects discussed are: (1) equations for compressible flow through turbomachines, (2) influence of axial velocity ratio on cascade performance, (3) three dimensional flow in transonic axial compressor blade rows, (4) prediction of turbulent shear layers in turbomachines, and (5) boundary layers in centrifugal compressors. For individual titles, see N75-11175 through N75-11190.

FLUID MECHANICS, ACOUSTICS, AND DESIGN OF TURBOMACHINERY, PART 2.
Lakshminarayana, ed., B.; Britsch, ed., W. R.; Gearhart, W. S.
Pennsylvania State Univ., University Park.
1974, 447p. refs.
NASA-SP-304-Pt-2; LC-72-600087-Pt-2
N75-11191
A conference was conducted to investigate various parameters involved in the design of turbomachinery. The acoustic properties of compressor rotors at subsonic speeds are described to show the sources of sound in fluid flows and sound radiation from the rotors. The design criteria for turbomachinery are examined to show impeller design methods, transonic compressor technology, and blade selection for an axial flow compressor. Specific applications of turbomachinery used as pumps for aerospace applications and turbomachinery for marine propulsion are described.

INVESTIGATION OF THE PHENOMENON OF ROTATING STALL
LeBot, Y.
Aero-Hydro-Elasticite, Summer Sch of Fluid Mech., Cycle of Conf., Ermenonville, Fr, Sep 4-8 1972, p.623-694

CALCULATION OF CASCADE PROFILES FROM THE VELOCITY DISTRIBUTION.
Lecomte, C.
Natl D'Etud et de Rech Aerosp., Chatillon, Fr.
ASME Pap. N 74-GT-70 for Meet. Mar 31-Apr 4 1974, p.6

COMPARISON OF THE EFFECT OF TWO DAMPER SIZES ON THE PERFORMANCE OF A LOW-SOLIDITY AXIAL-FLOW TRANSONIC COMPRESSOR ROTOR.
Lewis, G. W., Jr.; Urasek, D. C.
National Aeronautics and Space Administration. Lewis Research Center, Cleveland, Ohio
Apr 1972, 89p. refs.
NASA-TM-X-2536; E-6536
N72-22799
The experimental performance of a 20-inch diameter axial-flow transonic compressor rotor with small dampers is presented. The compressor rotor was tested earlier with large dampers which were twice in size, and comparisons of overall performance and radial distributions of selected flow and performance parameters are made. The rotor with small dampers experienced lower losses in the damper region which resulted in locally higher values of temperature rise efficiency and total pressure ratio. However, there was no appreciable effect on overall efficiency and pressure ratio. A greater stall margin was measured for the rotor with small dampers at design speed, but at 70 and 90 percent of design speed the rotor with large dampers had somewhat greater flow range.

SIMULATION OF VELOCITY PROFILES BY SHAPED GAUZE SCREENS.
Livesey, J. L.; Laws, E. M.
Univ. of Salford, Engl.
AIAA J V 11 N 2, Feb 1973, p.184-188

SHORT DURATION STUDIES OF TURBINE HEAT TRANSFER AND FILM COOLING EFFECTIVENESS.
Louis, J.F.; Demirjian, A. M.; Goulios, G. N.; Topping, R. F.; Wiedhopf, J. M.
ASME Pap. N 74-GT-131 for Meet. Mar 30-Apr 4 1974, p.15

INVESTIGATION OF ROTATING STALL IN AXIAL FLOW COMPRESSORS AND THE DEVELOPMENT OF A PROTOTYPE ROTATING STALL CONTROL SYSTEM.
Ludwig, G. R.; Nenni, J. P.; Arendt, R. H.
Calspan Corp. Buffalo, N.Y.
May 73, 280p.
MK-2932-A-13 AFAPL-TR-73-45
AD-762 297
The report summarizes the results of a three year program on rotating stall in axial flow compressors conducted at Calspan Corporation (formerly Cornell Aeronautical Laboratory). The work encompassed both experimental and theoretical investigations of rotating stall and the development of a prototype rotating stall control system. The experimental portion of the program included investigation of the effects of blade chord and solidity upon rotating stall properties and inception as well as an investigation of the effect of blade row rotation on blade row performance. In addition, an experiment to determine the stability of the flow through a blade row was conducted. A two-dimentional small-disturbance stability theory was developed to predict the inception of rotating stall. A single blade row and two blade row version of the theory were developed.

EXPERIMENTAL INVESTIGATION OF ADVANCED CONCEPTS TO INCREASE TURBINE BLADE LOADING. VOLUME 7 - PERFORMANCE EVALUATION OF MODIFIED JET-FLAP ROTOR BLADE.
Lueders, H. G.
General Motors Corp., Indianapolis, Ind. Allison Div.
Mar 71, 31p.
N71-20182

DIRECT PROBLEMS
Lumsdaine, E.; Fathy, A.
Univ. of Tenn., Knoxville
ASME Pap. N 74-GT-93 for Meet. Mar 31-Apr 4 1974, p.7

SUBCRITICAL AND SUPERCRITICAL COMPRESSIBLE SHOCK FLOW IN BLADE CASCADES
Luu, T. S.; Coulmy, G.; Grissa, A.
ATMA-Association Technique Maritime et Aeronautique. Paris (France)
1972, 33p.
N73-23378
Poisson type equations are used to study subcritical and supercritical flow about shockless blade cascades. Data cover pressure effects, flow velocity, fluid sliding on profiles, and perturbation. When blade geometry is considered for any angle of attack and for different Mach number values, pressure distribution and down stream flow condition may be determined. Whirlwind intensity and distribution on blade contours are also determined.

APPLICATION OF THE THEORY OF SIMULITUDE TO THE DESIGN OF CONTROL SYSTEMS FOR GAS TURBINE ENGINES
Lyubomudrov, Yu. V.
Foreign Technology Div. Wright-Patterson AFB, Ohio
25 May 73, 265p.
FTD-HC-23-1340-72
AD-764 683
The report discusses the questions related to optimum design using the theory of similitude of automatic control systems for aircraft gas turbine engines (gte) considered as objects with characteristics varying during flight. The equation of motion of the controlled object, expressed in terms of reduced variables, is unique for all conditions of its operation and may be represented as a graph of the 'dynamic characteristic of a Gte' that can be used as a basis to determine the required control laws. The selection of the optimal formula for the control system is done by using power complexes of Gte parameters. A foundation is provided for the principle, of designing a single control system that combines higher reliability of control in the entire range of flight conditions with reliability of control.

COMPUTER PROGRAM FOR PREDICTION OF AXIAL FLOW TURBINE PERFORMANCE
Macchi, E.
Naval Postgraduate School Monterey Calif.
Aug 70, 104p.
NPS-57Ma70081A
AD-713 116
The report presents a computer program for prediction of performance of single-stage axial turbines of given geometry. The three-dimensional method developed by Vavra is applied, taking account of streamline curvatures and slopes, as well as enthalpy and entropy gradients in the solutions of the equation of motion, and of boundary layer thicknesses in the continuity equation. A choice among five different loss correlation methods and two flow angles correlations is offered. Loss coefficients and flow angles are automatically calculated from blading geometry and actual flow conditions for every streamline, according to the selected correlation method.

AEROTHERMODYNAMIC FACTORS GOVERNING THE RESPONSE RATE OF GAS TURBINES
MacIsaac, B. D.; Saravanamuttoo, H. I. H.
Natl Res Counc Can Div Mech Eng Q Bull Rep DME/NAE N 3, 1974, p.23-25

COMPARISON OF ANALOG, DIGITAL AND HYBRID COMPUTING TECHNIQUES FOR SIMULATION OF GAS TURBINE PERFORMANCE.
MacIsaac, B.D.; Saravanamuttoo, H. I. H.
Carleton Univ., Ottawa, Ontario.
ASME Pap. 74-GT-127 for Meet. Mar 30-Apr 4 1974, p.12

CALCULATING TURBINE BLADE OSCILLATIONS ON ANALOG COMPUTERS
Magomaev, L. D.
Foreign Technology Div. Wright-Patterson AFB, Ohio
20 Feb 70, 15p.
FTD-HT-23-633-69
AD-704 013
The article states a method for calculating resonance frequencies, oscillation forms and tangential pressures originating in a blade, with the aid of an analog computer based on mathematical simulation procedures.

THE UNIQUENESS OF TURBOMACHINERY FLOW CALCULATIONS
Marsh, H.
Cambridge Univ. (England). Dept. of Engineering.
Feb 1971, 13p. refs.
CUED/A-Turbo/TR-24
N72-13264
When calculating the flow through turbomachines, it is assumed that on each cycle of iteration, there is only one solution for the flow pattern. The uniqueness of the solution obtained by the method of streamline curvature is examined and a set of Mach number conditions are derived which are sufficient to ensure that the flow pattern is unique. The Mach number limitations are the same as those which are necessary to avoid ambiguity in the matrix through-flow analysis. An alternative procedure is then described in which the solution for the flow pattern is always unique.

WALL BOUNDARY LAYERS IN TURBOMACHINES
Marsh, H.: Horlock, J. H.
Univ. of Durham, Engl.
J Mech Eng Sci V 14, N 6, Dec 1972, p.411-423

SECONDARY FLOW IN CASCADES: THE EFFECT OF AXIAL VELOCITY RATIO.
Marsh, H.
Univ. of Durham, Engl.
J Mech Eng Sci V 16, N 6, Dec 1974, p.402-407

XB-70 FLIGHT TEST DATA COMPARISONS WITH SIMULATION PREDICTIONS OF INLET UNSTART AND BUZZ
Martin, A. W.; Beaulieu, W. D.
North American Rockwell Corp., Los Angeles, Calif.
NASA Contract Rep. CR-1631, June 1970, p.28

HIGH PRESSURE STAGE OUTPUT OF MODERN TURBOREACTOR TURBINES
Martinat, P.
Societe Nationale d'Etude et de Construction de Moteurs d'Aviation. Paris (France)
1972, 27p.
N73-31695
An analysis was made of the factors influencing high pressure efficiency of turboreactor turbines. Data cover energy degradation as a function of technology, environment, blade cooling, and blade aerodynamics.

A REVIEW OF CURRENT AND PROJECTED ASPECTS OF TURBINE ENGINE PERFORMANCE EVALUATION
May, R. J. Jr.; Brownstein, B. J.; Przybylko, S. J.; McTasney, R. L.; Molisse, A. T.
Air Force Aero Propulsion Lab., Wright-Patterson, AFB, Ohio.
Feb 72, 117p.
AFAPL-TR-71-34
AD-744 587
The report is a compilation of four separate papers which, in total, represent a survey of several important aspects of turbine engine performance analysis. The first paper describes a digital computer method which has become a powerful tool for simulation of steady-state engine operation. A discussion of inlet airflow distortion is the topic of the second paper. It elaborates on a theory of rotating stall generation and a unique method which was encountered for determining how this distortion propagates through the compression components and produces surge. The third paper addresses the problems associated with the engine control

system and the techniques for computer simulation of transient engine operation. A discussion of the performance predictions of complex multimission aircraft weapon systems comprise the final section of this technical report.

MULTIHOLE COOLING FILM EFFECTIVENESS AND HEAT TRANSFER.
Mayle, R. E.; Camarata, F. J.
ASME Pap. N 74-HT-9 for Meet. Jul 15-17, 1974, p.10

INLET DYNAMICS AND COMPRESSOR SURGE
Mays, R. A.
J Aircraft U S N 4, Apr 1971, p.219-226

ANALYSIS OF INLET FLOW DISTORTION AND TURBULENCE EFFECTS ON COMPRESSOR STABILITY.
Melick, H. C.
Ltv Aerospace Corp., Dallas, Tex. Vought Systems Div.
31 Mar 73, 225p.
NASA-CR-114577; TR-2-57110/3R-3071
N73-21693
The effect of steady state circumferential total pressure distortion on the loss in compressor stall pressure ratio has been established by analytical techniques. Full scale engine and compressor/fan component test data were used to provide direct evaluation of the analysis. Specifically, since a circumferential total pressure distortion in an inlet system will result in unsteady flow in the coordinate system of the rotor blades, analysis of this type distortion must be performed from an unsteady aerodynamic point of view. By application of the fundamental aerothermodynamic laws to the inlet/compressor system, parameters important in the design of such a system for compatible operation have been identified. A time constant, directly related to the compressor rotor chord, was found to be significant, indicating compressor sensitivity to circumferential distortion is directly dependent on the rotor chord.

COMPUTER PROGRAM FOR THE PREDICTION OF DUCTED FAN PERFORMANCE
Mendenhall, M. R.; Spangler, S. B.
NASA CR-1495, Feb 1970.

EFFECTIVENESS AND HEAT TRANSFER WITH FULL-COVERAGE FILM COOLING.
Metzger, D. E.; Takeuchi, D. I.; Keunstler, P. A.
ASME Pap. N 73-GT-18 for Meet. Apr 8-12, 1973, p.5

ESTIMATION OF DEVIATION ANGLE FOR AXIAL-FLOW COMPRESSOR BLADE SECTIONS USING INVISCID-FLOW SOLUTIONS.
Miller, M. J.
National Aeronautics and Space Administration. Lewis Research Center, Cleveland, Ohio.
Mar 1974, 43p. refs.
NASA-TN-D-7549; E-7453
N74-17698
Development of a method of estimating deviation angles by analytical procedures was begun. Solutions for inviscid, irrotational flow in the blade-to-blade plane were obtained with a finite-difference calculation method. Deviation angles for a plane cascade with a rounded trailing edge were estimated by using the inviscid-flow solutions and three trailing-edge hypotheses. The estimated deviation angles were compared with existing experimental data over a range of incidence angles at inlet flow angles of 30 deg and 60 deg. The results indicate that deviation angles can be estimated accurately (within 1 deg) by using one of the three trailing-edge hypotheses, but only when pressure losses are low. A new trailing-edge hypotheses is presented which is suitable (for the cascade considered) for both low- and high loss operating points.

DEVIATION ANGLE ESTIMATION FOR AXIAL-FLOW COMPRESSORS USING INVISCID FLOW SOLUTIONS.
Miller, M. J.; Serovy, G. K.
J Eng Power Trans ASME V 97, Ser A N 2 , Apr 1975, p.163-172

LAMINAR BOUNDARY LAYER ON A ROTATING THREE-DIMENSIONAL BLADE.
Miyake, Y.; Fujita, S.
Osaka Univ., Jpn
J Fluid Mech V 65, Part 3, Sep 16, 1974, p.481-498

A STUDY OF LARGE SWALLOWING CAPACITY INWARD-FLOW RADIAL TURBINES.
Miyashita, T.; Sakakida, M.
IHI Eng Rev V7, N 2, May 1974, p.11-19

PREDICTION TECHNIQUES
Mokelke, H.
Motoren- Und Turbinen-Union, Munich, Ger.
AGARD Lect Ser N 72, 1974, for Meet., London, Engl., Nov 7-8 1974, Wright-Patterson AFB,Ohio, Nov 11-12 1974, and Nav Air Propul. Test Cent., Trenton, N.J., Nov 14-15 1974, Pap. 5, p.32

PERFORMANCE OF A SINGLE-STAGE AXIAL-FLOW TRANSONIC COMPRESSOR STAGE WITH A BLADE TIP SOLIDITY OF 1.7.
Moore, R. D.; Reid, L.
National Aeronautics and Space Administration. Lewis Research Center, Cleveland, Ohio.
Dec 1972, 113p. refs.
NASA-TM-X-2658; E-6730
N73-14983
The overall and blade-element performance of a transonic compressor stage is presented over the stable operating range at rotative speeds from 50 to 100 percent of design speed. Stage peak efficiency of 0.784 was obtained at a weight flow of 28.6 kilograms per second and a pressure ratio of 1.706. Stall margin at design speed was 11.4 percent. The peak efficiency being significantly less than design efficiency was attributed to: (1) the stator loss and the radial gradient of losses being much higher than design,

(2) the losses and blockages associated with the rotor part-span dampers not being incorporated into the design, and (3) mismatch of the rotor and stator blade elements.

OFF-DESIGN-POINT PERFORMANCES OF MULTI-SPOOL TURBO FAN ENGINES
Morita, M.; Sekine, S.
National Aerospace Lab., Tokyo (Japan)
1973, 25p. refs.
NAL-TR-347
N74-30242
A method for estimating the off design point performances of aircraft engines is presented. The method is applicable to the performance calculation of various forms of aircraft engines including single and twin spool turbojets and multi-spool turbofan engines. The steps to be followed in conducting the numerical analysis are outlined. The calculated results were compared with the flight test results of several turbofan engines with good correlation.

USE OF A POWERED MODEL FOR SUBSONIC NACELLE OPTIMIZATION
Motycka, D. L.; Disabato, V. J.; Andersen, L. Q.
Pratt & Whitney Aircraft, East Hartford, Conn
ASME Pap. 72-GT-14 for Meet. Mar 26-30 1972, p.9

NONLINEAR ANALYSIS OF ROTATING STALL
Nagano, S.; Takata, H.
Inst. Space Aeronaut Sci., Univ. Tokyo, Rep. 449, Apr 1970, p.119-197

CURVATURE EFFECTS ON A TURBINE BLADE COOLING FILM
Nicolas, J.; LeMeur, A.
ASME Pap. N 74-GT-156 for Meet. Mar 30-Apr 4 1974, p.16

EFFECTIVENESS OF FILM COOLING WITH THREE-DIMENSIONAL SLOT GEOMETRIES
Nina, M. N. R.; Whitelaw, J. H.
J Eng Power, Trans ASME V 93, Ser A N 4, Oct 1971, p.425-430

REAL-TIME SIMULATION OF JET ENGINES WITH A DIGITAL COMPUTER (1), FABRICATION AND CHARACTERISTICS OF THE SIMULATOR
Nishio, K.; Sugiyama, N.; Koshinuma, T.; Hashimoto, T.; Ohata, T.
Royal Aircraft Establishment Farnborough (England)
Dec 74, 51p.
RAE-Library Trans-1768 DRIC-BR-46783
AD-A007 319
Today the designers of engine control systems are required to provide the control systems of a newly-developed engine which is workable at the first test run. However, this means that both the dynamic and static characteristics of the new engine should be known in the early stage of development when the characteristics of each engine component are given. In such a case designers usually adopt a simulation technique. In this paper are reported the fabrication and performances of a real-time jet engine simulation test results with actual engine test data for the lift jet engine JR-100H.

SIMULATION OF TRIPLE-SPOOL TURBOFAN ENGINE.
Norvaisis, E. K.
Air Force Aero Propulsion Lab., Wright-Patterson AFB, Ohio
Apr 74, 156p.
AFAPL-TR-74-13
AD-784 771
This report describes a digital computer program entitled Trispl, trispl is a computer program that simulates steady-state design and off-design performance of triple-spool turbofan engines. The program has been formulated for an engine type with two core spools and one fan spool but can easily be modified for other engine types (two fan spools and one core spool, for example), the program, written in Fortran IV language, uses performance maps (in Block Data format) of the major engine components. Information on setting up the Block Data and input data is given in the report. Also included is a complete program listing with a description of each subroutine and sample results.

ACTUATOR DISK THEORY FOR INCOMPRESSIBLE HIGHLY ROTATING FLOWS.
Oates, G. C.
Univ. of Wash., Seattle
J Basic Eng., Trans ASME V 94, Ser D N 3, Sep 1972, p.613-621

THROUGHFLOW THEORY FOR TURBOMACHINES
Oates, G. C.; Knight, C. J.
Washington Univ., Seattle. Aerospace Research Lab.
Jun 1973, 120p. refs.
AFAPL-TR-73-61
N74-11602
Throughflow theory for flow in axial turbomachines is formulated in a way to allow consideration of the effects of variable hub and tip radii, the effects of free boundaries and the effects of compressibility. A procedure is developed for the very rapid calculation of incompressible flow through many blade rows in an annulus of constant hub and tip radii. An analysis formulation including the effects of variation in wall radii is also presented. A variational procedure for the numerical description of highly nonlinear flow field is developed for the description of incompressible flows through many blade rows in ducts of varying hub and tip radii. The associated computer program leads to rapid calculation of desired examples. Finally, a computer program is given to calculate the effects of compressibility on radial equilibrium flows.

EXPERIMENTAL PERFORMANCE IN ANNULAR CASCADE OF VARIABLE TRAILING- EDGE FLAP, AXIAL - FLOW COMPRESSOR INLET GUIDE VANES
Okiishi, T. H.; Junkhan, G. H.; Serovy, G. K.
ASME Pap. 70-GT-106 for Meeting May 24-28 1970, p.8

DIAGRAM OF OPERATIONAL CONDITIONS OF THE GT-100-750-2 LMZ GAS TURBINE PLANT
Ol'khovskii, G. G.; Ostrovskii, S. A.
All-union Heat Eng. Inst., USSR
Teploenergotika N 6, Jun 1973, p.25-30

PRESSURE WAVES PROPAGATION THROUGH BLADINGS IN AXIAL FLOW COMPRESSORS
Pandolfi, M.; Zannetti, L.
Politecnico di Torino (Italy). Ist. di Macchine e Motori per Aeromobili
May 74, 32p.
PUBL-161
N75-31014
Results of numerical computations of unsteady flow in axial turbocompressors are presented. The flow deflection by the blades and the axial propagation of pressure waves are considered in a model in which the actual blading is replaced by an infinite number of similar blades with zero thickness. This blading is in turn replaced by a field of forces chosen so as to satisfy the condition of tangency of the flow to the blades at each point. The equations of motion are detailed together with inlet and outlet boundary conditions, and conditions at the leading and trailing edges of the blades.

OPTIMIZATION OF HEAVILY LOADED COMPRESSOR BLADES BASED ON BOUNDARY LAYER THEORY.
Papailiou, K.
Von Karman Inst., for Fluid Dynamics. Rhoc Saint-genese (Belsium).
Sep 69, 2 7p.
VKI-TN-55
N70-17474

PROGRAM FOR THE DESIGN OF AN AXIAL COMPRESSOR STAGE BASED ON THE RADIAL EQUILIBRIUM EQUATIONS.
Papailiou, K. D.
Naval Postgraduate School Monterey Calif.
1971, 63p.
NPS-57PY719091A
AD-733 437
A computer program is presented to determine the three-dimensional flow conditions in an axial flow compressor stage. Entropy and energy gradients are taken into account as well as the radial shift and the curvatures of the axisymmetric stream surfaces. The program can be used at elevated Mach numbers since shock losses and compressibility effects are included. It represents an extension of work done for a research program to investigate the tip clearance effects in a three-stage compressor.

INCIDENCE EFFECTS AND CORNER STALL SUPPRESSION ON ROTOR BLADES IN COMPRESSORS.
Peacock, R. E.; Overli, J.
Can Congr of Appl Mech., 4th, Proc., Pap., Ec Polytech., Montreal, Quebec. May 28-Jun 1 1973, p.701-702

VIBRATIONS AND STABILITY OF TURBINE BLADES AT STALL.
Pigott, R.; Abel, J. M.
Westinghouse Electric Corp., Philadelphia, Pa.
J Eng Power, Trans ASME V 96, Ser A N 3, Jul 1974, p 201-208

PREDICTION AND MEASUREMENT OF PROPULSION SYSTEM PERFORMANCE.
Postlewaite, J.; Salemann, V.
Boeing Aerosp Co., Seattle, Wash.
J Eng Ind., Trans ASME V 96, Ser B N 3, Aug 1974, p.811-819

STUDY OF CASING TREATMENT STALL MARGIN IMPROVEMENT PHENOMENA.
Prince, D. C. Jr.; Wisler, D. C.; Hilvers, D. E.
ASME Pap. N 75-GT-60 for Meet. Mar 2-6 1975, p.12

COLD AIR STUDY OF THE EFFECT ON TURBINE STATOR BLADE AERODYNAMIC PERFORMANCE OF COOLANT EJECTION FROM VARIOUS TRAILING-EDGE SLOT GEOMETRIES. 1: EXPERIMENTAL RESULTS
Prust, H. W.; Bartlett, W. M.
National Aeronautics and Space Administration. Lewis Research Center. Cleveland, Ohio.
Mar 1974, 29p. refs.
NASA-TM-X-3000; E-7743
N74-17699
Trailing-edge slot configurations were investigated in a two-dimensional cascade of turbine stator blades. The trailing-edge slots were incorporated into blades with round trailing edges. The five blade configurations investigated included blades with two different trailing-edge thicknesses and four different slot widths. The results of the investigation showed that there was, in general, a significant increase in primary-air efficiency due to the coolant flow, the increase varying with slot configuration. For the five configurations tested, the average percent change in primary-air efficiency per percent coolant flow varied almost linearly from zero to about 1.4 percent over a range of coolant-to primary-air exit-velocity ratios between 0 and 1.2. However, for different configurations there was considerable deviation from the average values in the lower range of exit velocity ratios.

A COMPUTER SIMULATION OF THE TRANSIENT BEHAVIOR OF THE COMPRESSOR RESEARCH FACILITY INLET SYSTEM
Przybylko, S. J.
Air Force Aero Propulsion Lab Wright-Patterson AFB, Ohio
Jul 75, 138p.
AFAPL-TR-75-5
AD-A015 046
This report describes a digital computer simulation of the dynamics of the inlet and its control for a compressor research facility. The simulation uses lumped parameters and the Tustin method for the dynamic terms. The report discusses the convergency technique used for iteration and a Bode analysis performed to determine the gains for the proportional plus integral control used to position the inlet valves.

DECAY OF HIGHLY SKEWED FLOWS IN DUCTS.
Quinn, B.
Aerosp Res Lab., Wright-Patterson, AFB, Ohio
J Eng Power Trans ASME V97, Ser A N 1, Jan 1975, p.85-92

EFFECTIVENESS OF THREE-DIMENSIONAL FILM-COOLING SLOTS - 1. MEASUREMENTS.
Rastogi, A. K.; Whitelaw, J. H.
Int J Heat Mass Transfer V 16, N 9, Sep 1973, p.1665-1681

EXPERIMENTAL EVALUATION OF THE EFFECTS OF A BLUNT LEADING EDGE ON THE PERFORMANCE OF A TRANSONIC ROTOR.
Reid, L.; Urasek, D. C.
ASME Pap. N 73-GT-60 for Meet. Apr 8-12 1973, p.6

EFFECTS OF INCREASED LEADING-EDGE THICKNESS ON PERFORMANCE OF A TRANSONIC ROTOR BLADE
Reid, L.; Urasek, D. C.
National Aeronautics and Space Administration. Lewis Research Center, Cleveland, Ohio
Nov 1972, 57p. refs.
NASA-TN-D-7489; E-7076
N74-10025
A single-stage transonic compressor was tested with two rotor blade leading-edge configurations to investigate the effect of increased leading-edge thickness on the performance of a transonic blade row. The original rotor blade configuration was modified by cutting back the leading edge sufficiently to double the blade leading-edge thickness and thus the blade gap blockage in the tip region. At design speed this modification resulted in a decrease in rotor overall peak efficiency of four points. The major portion of this decrement in rotor overall peak efficiency was attributed to the flow conditions in the outer 30 percent of the blade span. At 70 and 90 percent of design speed, the modification had very little effect on rotor overall performance.

EFFECT OF REYNOLDS NUMBER AND LAMINAR SEPARATION ON AXIAL CASCADE PERFORMANCE.
Roberts, W. B.
Westinghouse Electric Corp., Sunnyvale, Calif.
ASME Pap. N 74-GT-68 for Meet. Mar 31-Apr 4 1974, p.13

DESIGN AND TEST OF A SMALL TWO-STAGE HIGH PRESSURE RATIO CENTRIFUGAL COMPRESSOR
Rodgers, C.; Langworthy, R. A.
ASME Pap. N 74-GT-137 for Meet. Mar 30-Apr 4 1974, p.13

PERFORMANCE OF A HIGHLY LOADED TWO STAGE AXIAL-FLOW FAN
Ruggeri, R. S.; Benser, W. A.
National Aeronautics and Space Administration. Lewis Research Center, Cleveland, Ohio
Aug 1974, 200p. refs.
NASA-TM-X-3076; E-7851
N74-31418
A two-stage axial-flow fan with a tip speed of 1450 ft/sec (442 m/sec) and an overall pressure ratio of 2.8 was designed, built, and tested. At design speed and pressure ratio, the measured flow matched the design value of 184.2 lbm/sec (83.55kg/sec). The adiabatic efficiency at the design operating point was 85.7 percent. The stall margin at design speed was 10 percent. A first-bending-mode flutter of the second-stage rotor blades was encountered near stall at speeds between 77 and 93 percent of design, and also at high pressure ratios at speeds above 105 percent of design. A 5 deg closed reset of the first-stage stator eliminated second-stage flutter for all but a narrow speed range near 90 percent of design.

ANALYSIS OF TURBOFAN ENGINE PERFORMANCE DETERIORATION AND PROPOSED FOLLOW-ON TESTS
Sallee, G. P.; Kruckenberg, H. D.; Toomey, E. H.
American Airlines, Inc., New York
1975, 130p.
NASA-CR-134769
N75-27039
Data and engine parts on in-service JT3D and JT8D engines were analyzed and documented relative to engine deterioration. It is concluded that the fan-compressor system of these engines contributes to the long term engine deterioration. An engine test and instrumentation plan was formulated for a proposed follow-on program. The goal of this program is to verify the above conclusion and to attempt to identify more precisely which components of the fan-compressor system are at fault.

CALCULATION OF THE FLOW THROUGH SUPERCRITICAL TURBINE CASCADES, WITH A VIEW TO DESIGNING BLADES WITH REDUCED SHOCK STRENGTH
Sampson, R. G.
Von Karman Inst. for Fluid Dynamics, Rhode Saint-Genese (Belgium)
Mar 1970, 89p. refs.
VKI-TN-57
N71-35410

A computer program has been produced which calculates the transonic and supersonic flow regions in a supercritical turbine cascade with a well defined throat. The computer results have been used to demonstrate the effects an expansion on the pressure surface, and a recompression on the suction surface, have on the performance of an existing design.

MATHEMATICAL MODELLING OF GAS TURBINE SUPERCHARGING IN MULTICYLINDER FOUR-CYCLE ENGINES
Samsonov, L. A.
Army Foreign Science and Technology Center, Washington, D.C.
9 Feb 70, 15p.
FSTC-HT-23-358-70
AD-713 873
The principle features are presented of a method of mathematical modelling of the operation of a supercharging system for multicylinder four-cycle engines developed; a comparison is presented of the results of modelling of the operation of the supercharger for engine 40hn15.5/20.5 using the "Ural-2" computer with experimental data. Examples are presented of the investigation of the influence of design of the system on its parameters.

SIMULATION OF GAS TURBINE DYNAMIC PERFORMANCE
Saravanamuttoo, H. I. H.; Fawke, A. J.
Univ. of Bristol, England.
ASME Pap. 70-GT-23 for Meet. May 24-28 1970, p.8

USE OF A HYBRID COMPUTER IN THE OPTIMIZATION OF GAS TURBINE CONTROL PARAMETERS.
Saravanamuttoo, H. I. H.; MacIsaac, B. D.
Carleton Univ., Ottawa, Ontario.
ASME Pap. N 73-GT-13 for Meet. Apr 8-12 1973, p.8

ROTOR DESIGN TO ATTENUATE FLOW DISTORTION - 1.
Savell, C. T.; Wells, W. R.
GE, Evendale, Ohio
J Eng Power Trans ASME V 97, Ser A N 1, Jan 1975, p.11-20

ROTOR DESIGN TO ATTENUATE FLOW DISTORTION - 2. AN UNSTEADY THIN AIRFOIL CASCADE ANALYSIS.
Savell, C. T.; Wells, W. R.
GE, Schenectady, NY
J Eng Power Trans ASME V 97, Ser A N 1, Jan 1975, p.37-46

HYBRID SIMULATION OF A SINGLE-SHAFT GAS TURBINE WATER PUMP DRIVE.
Schatborn, I. W.
Werkspoor-Amsterdam, Neth.
ASME Paper N 73-GT-72 for Meet. Apr 8-12 1973, p.16

GENERALIZED DYNAMIC ENGINE SIMULATION TECHNIQUES FOR THE DIGITAL COMPUTER.
Sellers, J.; Teren, F.
National Aeronautics and Space Administration. Lewis Research Center. Cleveland, Ohio
1974, 25p.
NASA-TM-X-71552; E-7968
N74-27294
Recently advanced simulation techniques have been developed for the digital computer and used as the basis for development of a generalized dynamic engine simulation computer program called Dyngen, this computer program can analyze the steady state and dynamic performance of many kinds of aircraft gas turbine engines. Without changes to the basic program Dyngen can analyze one- or two-spool turbofan engines. The user must supply appropriate component performance maps and design-point information. Examples are presented to illustrate the capabilities of Dyngen in the steady state and dynamic modes of operation. The analytical techniques used in Dyngen are briefly discussed, and its accuracy is compared with a comparable simulation using the hybrid computer. The impact of Dyngen and similar all-digital programs on future engine simulation philosophy is also discussed.

DYNGEN: A PROGRAM FOR CALCULATING STEADY-STATE AND TRANSIENT PERFORMANCE OF TURBOJET AND TURBOFAN ENGINES
Sellers, J. F.; Daniele, C. J.
National Aeronautics and Space Administration. Lewis Research Center, Cleveland, Ohio
Apr 1975, 202p. refs.
NASA-TN-D-7901; E-8111
N75-25620
The DYNGEN, a digital computer program for analyzing the steady state and transient performance of turbojet and turbofan engines, is described. The DYNGEN is based on earlier computer codes (SMOTE, GENENG, and GENENG 2) which are capable of calculating the steady state performance of turbojet and turbofan engines at design and off-design operating conditions. The DYNGEN has the combined capabilities of GENENG and GENENG 2 for calculating steady state performance; to these the further capability for calculating transient performance was added. The DYNGEN can be used to analyze one- and two-spool turbojet engines or two- and three-spool turbofan engines without modification to the basic program. A modified Euler method is used by DYNGEN to solve the differential equations which model the dynamics of the engine. This new method frees the programmer from having to minimize the number of equations which require iterative solution. As a result, some of the approximations normally used in transient engine simulations can be eliminated. This tends to produce better agreement when answers are compared with those from purely steady state simulations. The modified Euler method also permits the user to specify large time steps (about 0.10 sec) to be used in the solution of the differential equations. This saves computer execution time when long transients are run. Examples of the use of the program are included, and program results are compared with those from an existing hybrid-computer simulation of a two-spool turbofan.

UTILIZATION OF CASCADE DATA IN AXIAL - FLOW COMPRESSOR DESIGN AND ANALYSIS - A CRITICAL REVIEW
Serovy, G. K.
ASME Pap. 70-GT-108 for Meeting May 24-28 1970, p. 12

PREDICTION OF OVERALL AND BLADE-ELEMENT PERFORMANCE FOR AXIAL-FLOW PUMP CONFIGURATIONS.
Serovy, G. K.; Kavanagh, P.; Okiishi, T. H.; Miller, M. J.
Iowa State Univ., of Science and Technology, Ames
Aug 73, 248p.
NASA-CR-2301; ISU-ERI-Ames-72322
N73-32189
A method and a digital computer program for prediction of the distributions of fluid velocity and properties in axial flow pump configurations are described and evaluated. The method uses the blade-element flow model and an iterative numerical solution of the radial equilibrium and continuity conditions. Correlated experimental results are used to generate alternative methods for estimating blade-element turning and loss characteristics. Detailed descriptions of the computer program are included, with example input and typical computed results.

DESIGN OF COMPRESSORS. AERODYNAMIC CALCULATIONS
Sherstyuk, A. N.
Foreign Technology Div Wright-Patterson AFB, Ohio
26 May 71; 331p
FTD-HC-23-781-70
AD-727 191
The book sets forth the foundations of the theory and design of axial compressors. Particular attention is devoted to compressors of stationary installations. The book is intended for engineers who design axial compressors. In addition, the book can be used by students in higher technical training institutions.

PROBLEM OF THE OPTIMIZATION OF THE GEOMETRIC PARAMETERS OF A TURBINE STAGE DURING THE TWISTING OF THE FIXED AND MOVING BLADES
Shkvar, A. Ya.
Foreign Technology Div. Wright-Patterson AFB, Ohio
2 Apr 74, 15p.
FTD-HT-23-566-74
AD-778 521
The perfecting of the flow part of contemporary turbine stages is provided mainly by two methods: the use of vane cascades with great aerodynamic characteristics, and mutual agreement of the geometric parameters of the cascade in order to guarantee optimum efficiency of modes for their joint operation. Especially significant is the importance of the latter factor in stages with relatively long blades where for an increase in the efficiency of a stage different forms of twists of fixed and moving blades are used.

FLOW INTO A TRANSONIC COMPRESSOR ROTOR
Shreeve, R. P.
Naval Postgraduate School Monterey Calif
Aug 74, 33p.
NPS-57F74081
AD-A011 496
An analytical representation is given for the axisymmetric 2-dimensional flow field at the face of a transonic rotor. Streamlines computed for irrotational inviscid flow are closely approximated using a curve-fitting technique, which is described, and boundary layer blockage is then introduced. With the inlet flow so represented, the conditions relative to the rotor blading of the TRANSX compressor are calculated as a function of the radius and graphed as a function of flow rate and rotor speed. The purpose of this report is to both document calculations carried out for the TRANSX compressor and to record useful analytical techniques and short programs developed.

RESEARCH ON THE FLUTTER OF AXIAL-TURBOMACHINE BLADING
Sisto, F.; Perumal, P. V. K.
Stevens Inst of Tech Hoboken N J Dept of Mechanical Engineering
2 May 73, 39p.
ME-RT-73003
AD-760 354
An analytical method for predicting the perturbed aerodynamic reactions of a harmonically oscillating flat plate airfoil with time dependent point of separation is presented. It is shown that this method in conjunction with an empirical knowledge of the time history of the separation point can predict stall flutter. Numerical results are presented and compared with existing theoretical and experimental results.

RESEARCH ON THE FLUTTER OF AXIAL-TURBOMACHINE BLADING
Sisto, F.; Ho Ni, R.
Stevens Inst of Tech Hoboken N J Dept. of Mechanical Engineering
May 74, 31p.
ME-RT-740008
AD-780 467
The 'time marching' technique is successfully applied to the numerical computation of the nonstationary aerodynamics of a flat plate cascade for compressible flow of either subsonic or supersonic nature. The unsteady perturbation amplitudes of fluid properties are used as the dependent variables so that the computational domain can be reduced to a two-dimensional channel guided by two adjacent blades for any inter-blade phase angle. A new method of handling the boundary condition is developed with which the order of accuracy for the boundary points will be the same as for the interior points. The wake region behind the trailing edge of each blade in cascade is treated as a 'slip plane' as in two dimensional steady state supersonic flow. Results are in good agreement with existing solutions.

USE OF ANALYTIC SURFACES FOR THE DESIGN OF CENTRIFUGAL IMPELLERS BY COMPUTER GRAPHICS
Smith, D. J. L.; Merryweather, H.
Natl Gas Turbine Establ., Hants, Engl.
Int J Numer Methods Eng V 7, N 2, 1973, p.137-154

A STAGE-STACKING SIMULATION OF AXIAL FLOW COMPRESSORS WITH VARIABLE GEOMETRY
Southwick, R. D.
Aeronautical Systems Div Wright-Patterson AFB, Ohio
Oct 74, 93p.
ASD-TR-74-38
AD-A006 407
Since most high performance axial flow compressors in development today have variable geometry, it is highly desirable to have the capability to evaluate various stator schedules as to their effect on compressor performance using computer simulation. The computer program covered in this report uses the 'stage-stacking' method of analyzing compressor performance and uses input derived from test data in lieu of detailed design information. It can accommodate compressors with up to 15 stages, 10 of which can have variable stators. In addition to providing individual stage performance and overall compressor performance in a printout format, the program also contains a plotting routine for displaying the overall compressor map in graphical form.

LIFE PREDICTION OF TURBINE COMPONENTS ON-GOING STUDIES AT THE NASA LEWIS RESEARCH CENTER
Spera, D. A.; Grisaffe; S. J.
National Aeronautics and Space Administration, Lewis Research Center
Jan 73, 71p.
NASA-TM-X-2664; E-7158
N73-15925
An overview is presented of the many studies at NASA-Lewis that form the turbine component life prediction program. This program has three phases: (1) development of life prediction methods for major failure modes through materials studies. (2) evaluation and improvement of these methods through a variety of burner rig studies on simulated components in research engines and advanced rigs. These three phases form a cooperative, interdisciplinary program. A bibliography of Lewis publications on fatigue, oxidation and coatings, and turbine engine alloys is included.

INCIDENCE LOSS FOR A CORE TURBINE ROTOR BLADE IN A TWO-DIMENSIONAL CASCADE
Stabe, R. G.; Kline, J. F.
National Aeronautics and Space Administration, Lewis Research Center. Cleveland, Ohio.
Apr 74, 13p.
NASA-TM-X-3047; E-7835
N74-21397
The effect of incidence angle on the aerodynamic performance of an uncooled core turbine rotor blade was investigated experimentally in a two-dimensional cascade. The cascade test covered a range of incidence angles from minus 15 deg to 15 deg in 5-degree increments and a range of pressure ratios corresponding to ideal exit critical velocity ratios of 0.6 to 0.95. The principal measurements were blade-surface static pressures and cross-channel surveys of exit total pressure, static pressure, and flow angle. The results of the investigation include blade-surface velocity distribution and overall performance in terms of weight flow and loss for the range of incidence angles and exit velocity ratios investigated. The measured losses are also compared with two common methods of predicting incidence loss.

RESEARCH TRENDS IN TURBINE AERODYNAMICS
Stewart, W. L.; Glassman, A. J.
National Aeronautics and Space Administration. Lewis Research Center, Cleveland, Ohio
1972, 7p.
NASA-TM-X-68016; E-6814
N72-18782
Recent trends in turbine aerodynamics are summarized. Areas discussed include cooled turbine aerodynamics, high work factor turbines, pneumatic variable geometry, and computer analysis.

THE EFFECTS OF UNIFORM PROFILE CHANGES ON THE DEGREE OF EFFICIENCY, THE INCLINE, AND THE MASS FLOW IN MULTIPLE STEP AXIAL TURBINES.
Stobbe, H.
Technische Hochschule Hannover (West Germany)
1969, 219p
N70-39449

TWO-DIMENSIONAL METHOD FOR CALCULATING SEPARATED FLOW IN A CENTRIFUGAL IMPELLER
Sturge, D. P.; Cumpsty, N. A.
Univ of Cambridge, Engl
ASME Pap N 75-FE-6 for Meet. May 5-7 1975, p.9

AERODYNAMIC EFFECTS OF TIP CLEARANCE IN A LIFTING LINE IN A NON-UNIFORM FLOW.
Sugiyama, Y.
Max-Planck-Institut fuer Stroemungsforschung. Goettingen (West Germany)
Apr 73, 36p.
Rept-4/1973
N73-28738
The aerodynamic performance of a lightly loaded lifting line which lies in a nonuniform flow of a perfect fluid through a channel and which is interfered upon by channel walls near tips of the line, were calculated. The induced velocity potential due to the lifting line or the channel walls were calculated. By applying the relation in the lifting line theory, which implies that the effective angle of attack of a blade is equal to the sum of the geometrical angle and the downwash angle, the lift of the lifting line was connected with induced velocities obtained from the velocity potentials. Expressions were derived, from which the lift and the induced drag acting on the lifting line can be calculated. The effects of tip clearances or the nonuniformity of the flow on the aerodynamic performances of the lifting line are demonstrated numerically.

SURGE LIMIT ON MULTI-STAGE AXIAL COMPRESSORS
Suter, P.; Spati, H.
Turboforum N 2, Jul 1972, 85-93p.

RESEARCH TURBINE FOR HIGH-TEMPERATURE CORE ENGINE APPLICATION. 2: EFFECT OF ROTOR TIP CLEARANCE ON OVERALL PERFORMANCE.
Szanca, E. M.; Behning, F. P.; Schum, H. J.
National Aeronautics and Space Administration. Lewis Research Center, Cleveland, Ohio
Apr 1974, 30p.
N74-19644
NASA-TN-D-7639; E-7728
A 25.4-cm (10-in) tip diameter turbine was tested to determine the effect of rotor radial tip clearance on turbine overall performance. The test turbine was a half-scale model of a 50.8-cm (20-in.-) diameter research turbine designed for high-temperature core engine application. The test turbine was fabricated with solid vanes and blades with no provision for cooling air and tested at much reduced inlet conditions. The tests were run at design speed over a range of pressure ratios for three different rotor clearances ranging from 2.3 to 6.7 percent of the annular blade passage height. The results obtained are compared to the results obtained with three other turbines of varying amounts of reaction.

REAL-TIME SIMULATION OF F100-PW-100 TURBOFAN ENGINE USING THE HYBRID COMPUTER
Szuch, J. R.; Seldner, K.
National Aeronautics and Space Administration. Lewis Research Center, Cleveland, Ohio
Aug 1975, 80p.
NASA-TM-X-3261; E-8136
N75-28068
A real-time hybrid computer simulation of the F100-PW-100 augmented turbofan is presented. The digital portion of the hybrid computer is used to perform the bivariate function generation associated with modelling the performance of the engine's rotating components. The remaining calculations are performed on the analog computer. Steady state simulation data along with sea level, static, transient data are presented to show that the real-time simulation matches baseline digital simulation results over a wide range of power settings and flight conditions. Steady state simulation data are compared with sea level, experimental data to show that the real-time hybrid and baseline digital simulations do adequately predict the performance of the actual engine. FORTRAN listings and analog patching diagrams are provided.

SIMULATION OF ENVIRONMENTAL SOLID-PARTICLES TRAJECTORIES AND VELOCITIES THROUGH AN AXIAL FLOW COMPRESSOR STAGE, AND THE PRESSURE DISTRIBUTION ON BLADES.
Tabakoff, W.; Hamed, A.; Hussein, M. E.
Cincinnati Univ Ohio
1971, 32p.
AD-725 596
An experimental investigation is reported of the trajectories and velocities of solid particles suspended in a fluid passing through an axial flow compressor stage. Such investigation is of importance to the study of erosion damage sustained by the blade. Two test facilities were used for this study: a subsonic cascade wind tunnel for compressible flow and a water table for incompressible flow. From the test technique it would appear that the present existing theoretical analysis for particle trajectories through a compressor stage is questionable. The wind tunnel test simulation is preferred for prediction particle trajectories.

AN ANALYTICAL STUDY OF FLOW LOSSES THROUGH A TWO-DIMENSIONAL TURBINE CASCADE WITH BOUNDARY LAYER INJECTION.
Tabakoff, W.; Earley, R.
Cincinnati Univ Ohio Dept of Aerospace Engineering
Apr 72, 52p.
72-26
AD-744 535
A method for determining the performance of a two-dimensional turbine cascade with boundary layer injection is developed using existing incompressible boundary layer approximate solutions with a new formulation for the injection. The overall cascade loss includes friction and wake mixing losses. The results of the analysis are compared with experimentally obtained data as a check of the validity of the new analytical method.

NONLINEAR ANALYSIS OF ROTATING STALL
Takata, H.; Nagano, S.
J Eng Power, Trans ASME V 94, Ser A N 4, Oct 1972, p.279-293

STUDY ON THE MECHANISM OF STALL MARGIN IMPROVEMENT OF CASING TREATMENT
Takata, H.; Tsukuda, Y.
ASME Pap. N 75-GT-13 for Meet. Mar 2-6 1975, p.16

ON THE PARTIAL FLOW RATE PERFORMANCE OF AXIAL-FLOW COMPRESSOR AND ROTATING STALL - 1. INFLUENCES OF HUB-TIP RATIO AND STATORS
Tanaka, S.; Murata, S.
Bull JSME V 18, N 117, Mar 1975, p.256-263

ON THE PARTIAL FLOW RATE PERFORMANCE OF AXIAL-FLOW COMPRESSOR AND ROTATING STALL - 2. INFLUENCES OF IMPELLER LOAD AND A STUDY OF THE MECHANISM OF UNSTABLE PERFORMANCES.
Tanaka, S.; Murata, S.
Bull JSME V18, N 117, Mar 1975, p.264-271

IMPROVED METHOD OF AND APPARATUS FOR PREVENTING COMPRESSOR STALL IN A GAS TURBINE ENGINE
Thomson, F.C.
Department of the Navy Washington D.C.
13 Aug 74, 11p.
PAT-APPL-497 017
The patent application relates to a method of, and apparatus for, detecting air distortion at the inlet of a gas turbine engine and adjusting fuel flow to anticipate and prevent compressor stall. A plurality of pressure taps are arranged in a spaced relation around the periphery of the engine inlet. A distortion detector senses the differential pressure between the instantaneous pressure at each pressure tap and the ambient pressure of a reference pressure chamber which communicates with the plurality of pressure taps. At a predetermined pressure differential the distortion detector activates a solenoid operated fuel bypass valve which reduces fuel flow to the fuel nozzles of the gas turbine engine.

PREDICTION OF FLOW OUTLET ANGLE IN BLADE ROWS WITH CONICAL STREAM SURFACES
Traupel, W.
Swiss Federal Inst of Tech., Zurich, Switz
ASME Paper N 73-GT-32 for Meet. Apr 8-12 1973, p.8

DYNAMIC STABILITY OF WING-MOUNTED ENGINE INSTALLATIONS FOR TURBOPROP-POWERED AIRCRAFT
Trne, D. J.; Alderson, R. G.; Harvey, J. W.; Mason, D. R.
Airesearch Manuf Co of Ariz, Phoenix
J Eng Power Trans ASME V 97, Ser A N 2, Apr 1975, p.275-282

INFLUENCE OF TRANSIENT CONDITIONS ON OVERALL SERVICE LIFE OF TURBINE BLADES
Tretyachenko, G. N.
Techtran Corp., Glen Burnie, Md.
Sep 73, 8p.
NASA-TT-F-15113
N73-31700
It is shown that in spite of their relatively short duration, transient modes of operation of the type occurring during takeoff, landing, and engine tests have a tremendous effect on the service life of turbine blades. In view of this, it is suggested to carry out accelerated tests by determining the time to failure at steady modes of operation on the basis of data obtained with cylindrical samples, and at transient modes of operation, on the basis of tests performed with actual blades under simulated conditions.

GAS-DYNAMIC VERIFICATION CALCULATION OF AXIAL-FLOW MULTISTAGE COMPRESSORS ON EDC
Tunakov, A. P.; Ibragimov, S. G.
Foreign Technology Div Wright-Patterson AFB, Ohio
1 Nov 72, 21p.
FTD-MT-24-1501-72
AD-753 519
An algorithm for testing the gas dynamic design of axial-flow compressors, particularly in the final adjustment phase, is proposed. Particularly well suited for calculating the parameters of blade rings, the procedure is suitable for use in the design phase if data on the loss coefficients in empirical formulas have been established.

THE EFFECT OF TECHNOLOGICAL TOLERANCES ON GAS TURBINE PARAMETERS
Tunakov, A. E.; Rzhavin, Yu, A.
Foreign Technology Div Wright-Patterson AFB, Ohio
27 Nov 74, 16p.
FTD-HC-23-1487-74
AD/A004 349
A mathematical model of a turbine stage cell is devised such that for any operational condition all the basic parameters of the turbine can be determined. Solution to the problem was carried out with the usually employed assumptions of turbomachine theory.

EXPERIMENTAL STUDY OF AN INTERNAL AIR-COOLED GAS TURBINE BLADE
Tyryshkin, V. G.; Gruntfest, M. I.; Mikhailova, V. A.
Foreign Technology Div Wright-Patterson AFB, Ohio
26 Jun 75, 18p.
FTD-ID(RS)I-1479-75
AD-A014 523
Aviation gas turbines are making more use of blades internally cooled by air taken from the cycle. Such blades reduce temperatures ahead of the turbine, increasing specific power and decreasing fuel flow, but involve energy losses and decrease of efficiency. A study is reported of one construction of a moving blade cooled by forced convection, in which the cooling air escapes from the blade into the interior of the turbine.

PERFORMANCE OF TRANSONIC FAN STAGE WITH WEIGHT FLOW PER UNIT ANNULUS AREA OF 208 KILOGRAMS PER SECOND PER SQUARE METER (42.6 (LB/SEC)/SQ FT)
Urasek, D. C.; Kovich, G.; Moore, R. D.
National Aeronautics and Space Administration. Lewis Research Center, Cleveland, Ohio
Nov 1973, 98p.
NASA-TM-X-2903; E-7229
N74-10022
Performance was obtained for a 50-cm-diameter compressor designed for a high weight flow per unit annulus area of 208 (kg/sec)/sq m. Peak efficiency values of 0.83 and 0.79 were obtained for the rotor and stage, respectively. The stall margin for the stage was 23 percent, based on equivalent weight flow and total-pressure ratio at peak efficiency and stall.

FORTRAN PROGRAM FOR CALCULATING VELOCITIES IN THE MERIDIONAL PLANE OF A TURBOMACHINE I CENTRIFUGAL COMPRESSOR
Vanco, M. R.

National Aeronautics and Space Administration. Lewis Research Center. Cleveland, Ohio
Mar 72, 57p.
NASA-TN-D-6701; E-6592
N72-17988
The program will determine the velocities in the meridional plane of a backward-swept impeller, a radial impeller, and a vaned diffuser. The velocity gradient equation with the assumption of a hub-to-shroud mean stream surface is solved along arbitrary quasi-orthogonals in the meridional plane. These quasi-orthogonals are fixed straight lines.

APPROXIMATION OF THE BLADE SURFACE AND BOUNDARY SURFACES BY A TRIGONOMETRIC POLYNOMIAL
Vavilov, G. A.; Garifov, R. K.; Korabel'nikov, V. Z.; Tarasov, V. N.
Energomashinostroenie N 12, Dec 1974, p.12-14

CALCULATING METHOD FOR MULTI-STAGE AXIAL COMPRESSORS WITH IMPLUSE BLADINGS AND CONSTANT TIP DIAMETER
Vavra, M. H.
Naval Postgraduate School Monterey Calif.
Mar 74, 73p.
NPS-57VA74031
AD-778 808
The report gives an approximate calculating method for the design point performance of multi-stage axial compressors with impulse-type bladings and constant tip diameters. Computing programs for Monroe-1180 programmable calculators are presented to establish the compressor performance and the blading parameters for arbitrary conditions with minimum effort. The report was prepared to permit evaluations of the applicability of such compressors in advanced propulsion units for air-superiority aircraft, or in light-weight lift engines for military VTOL aircraft.

PROCEEDINGS OF THE WORKSHOP ON FLOW IN TURBOMACHINES
Vavra, M. H.; Papailiou, K. D.; Woods, J. R. Jr.
Naval Postgraduate School Monterey Calif
16 Nov 71, 439p.
NPS-57VA71111A
AD-735 021
Contents: Non-steady phenomena in transonic and supersonic flows and possible methods of solution; Some recent developments in the numerical analysis and simulation of fluid turbulence; A review of the history of boundary layer calculation methods and the present state of the art; New measuring and flow-visualization techniques; Applicability of cascade test data to design methods; Radial equilibrium across a normal shock in an axial rotor; Three-dimensional, inviscid flow analysis in turbomachinery; Application of results of research to engine design problems; Loss evaluation methods in axial-flow compressors; The turbulence structural hypothesis and loss coefficient predictions; Loss correlations and off-design performance predictions; Flutter; Noise; Stall and surge; and turbine blade cooling.

PREDICTION OF BOUNDARY LAYER DEVELOPMENT ON AXIAL-FLOW TURBOMACHINE BLADES
Walker, G. J.
Instn of Engrs., Australia-3rd Australiasian Conference on Hydraulics & Fluid Mechanics-Proc, Nov 25-29 1968
Paper 259, p.6

MATCHING OF HIGH-OUTPUT TURBOCHARGED ENGINES FOR MAXIMUM TORQUE BACKUP AND EMISSION REDUCTION BASED ON THE USE OF VARIABLE GEOMETRY COMPRESSORS AND TURBINES
Wallace, F. J.; Sivakuraman, K.
SAE Prepr N 740738 for Meet. Sep 9-12 1974, p.18

CALCULATED PERFORMANCE MAP OF A 4 1/2-STAGE 15.0 CENTIMETER (5.9 INCH) MEAN DIAMETER TURBINE DESIGNED FOR A TURBOFAN SIMULATOR.
Wasserbauer, C. A.
Cincinnati Univ., Ohio
Jun 73, 14p.
NASA-TM-X-2822; E-7403
N73-25822
The overall performance of an existing high-ratio turbine is calculated analytically over a range of speed and pressure ratio in order to determine its capability for other applications. The analytical performance covers a speed range from 50 to 120 percent of design and a pressure-ratio range from 5.0 to 35.0. The turbine was designed for a 50.8 centimeter (20.0 in.) tip diameter turbofan results are compared with the experimental turbine data obtained from testing three fan configurations with the turbofan simulator in air. The comparison indicates good agreement over the range of speeds and pressure ratios covered by the experimental data.

FORTRAN PROGRAM FOR PREDICTING OFF-DESIGN PERFORMANCE OF RADIAL-INFLOW TURBINES
Wasserbauer, C. A.; Glassman, A. J.
National Aeronautics and Space Administration. Lewis Research Center. Cleveland, Ohio
Sep 75, 55p
NASA-TN-D-8063; E-8368
N75-32118
The FORTRAN IV program uses a one-dimensional solution of flow conditions through the turbine along the mean streamline. The program inputs needed are the design-point requirements and turbine geometry. The output includes performance and velocity-diagram parameters over a range of speed and pressure ratio. Computed performance is compared with the experimental data from two radial-inflow turbines and with the performance calculated by a previous computer program. The flow equations, program listing, and input and output for a sample problem are given.

EXPERIMENTAL INVESTIGATION OF THE EFFECTS OF TIP CLEARANCE AND END LOSSES ON AXIAL COMPRESSOR PERFORMANCE
Welch, J. K.
Naval Postgraduate School Monterey Calif.
Jun 73, 135p.
AD-767 656
The objective of the study was to determine by experimental means the rotor efficiencies at different radii between hub and tip of a single stage axial compressor at its design point, to show the influence of tip clearance and end losses. Procedures for calibration and application of pressure probes to survey the flow in the compressor were established and programs were written to analyze the measured data. Recommendations are made for improvements of the data reduction method, which should precede experiments involving controlled changes in the blade tip clearances.

CALCULATION OF SUPERSONIC COMPRESSOR LOSSES
Wells, W. R.; Tabakoff, W.
Cincinnati Univ Ohio Dept of Aerospace Engineering
26 Oct 70, 6p.
AROD-T-4:33-F
AD-728 077
The purpose of the analysis is the development of a simple realistic means to compute the adiabatic efficiency of a supersonic compressor.

EXPERIMENTAL DETERMINATION OF TURNING ANGLE AND LOSSES OF AXIAL COMPRESSOR INLET GUIDE VANES
Wheeler, W. R.
Naval Postgraduate School Monterey Calif.
Dec 72, 47p.
AD-757 250
The investigation experimentally determined the minimum loss incidence angle, deviation angle, and total-pressure loss coefficient for a cascade with airfoil-type blade profiles used as inlet guide vanes for an axial-flow compressor with an equivalent camber angle of 37.6 degrees and unit solidity. The experimental values were compared with values predicted using correlations based on compressor cascade tests.

COLD-AIR INVESTIGATION OF A 3 1/2- STAGE FAN-DRIVE TURBINE WITH A STAGE LOADING FACTOR OF 4 DESIGNED FOR AN INTEGRAL LIFT ENGINE. 1: TURBINE DESIGN AND PERFORMANCE OF FIRST STAGE
Whitney, W. J.; Schum, H. J.; Behning, F. P.
National Aeronautics and Space Administration. Lewis Research Center. Cleveland, Ohio
Oct 75, 39p.
NASA-TM-X-3289; E-8354
N75-33057
The design of the 3 1/2-stage turbine is described, and the cold-air performance of the first stage, modified for axial inlet conditions, is presented. The performance of the modified single-stage turbine and of two contemporary high-stage-loading-factor turbines is compared with that estimated with a reference prediction method.

OPTIMIZATION OF THE RESPONSE OF FRICTIONALLY DAMPED BEAM TYPE STRUCTURES WITH REFERENCE TO GAS TURBINE COMPRESSOR BLADING.
Williams, E. J.; Earles, S. W. E.
Univ of Nottingham, University Park, Engl.
J Eng Ind., Trans ASME V 96, Ser B N 2, May 1974, p.471-476

TRANSPIRATION COOLING. PART I. ANALYTICAL MODEL.
Winget, L.; Han, L. S.
Ohio State Univ. Research Foundation Columbus
Dec 72, 420p.
AFAPL-TR-73-11-PT-1
AD-759 190
An analytical method is presented for investigation of transpiration cooling and the downstream cooling effects beyond the point of discontinuous blowing. The method presented is restricted to the laminar regime. An integral method which satisfies the first two compatibility conditions at the wall is developed for both the velocity and thermal boundary layers. An exponential function is incorporated for both the velocity and temperature fields. The boundary layer definition is extended through the position of discontinuous blowing to the point of separation. The wall temperature is assumed constant in the transpiration regime and is allowed to vary downstream from the permeable wall to evaluate the effects of downstream cooling. Asymptotic expressions are given for both large blowing and suction velocities.

TRANSPIRATION COOLING. PART II. EXPERIMENTAL STUDY OF WAKE EFFECTS.
Winget, L. E.; Han, L. S.
Ohio State Univ Research Foundation Columbus
Mar 73, 121p.
AFAPL-TR-73-11-Pt-2
AD-759 191
An experimental analysis is presented for the investigation of the stator wake effects on the rotor blades which are immediately downstream. A two-dimensional test apparatus is used to simulate the stator and rotor blades. The test section is 2 x 2 feet. The chord length of the test blades is 7 inches. All Reynolds number range is from 100,000 to 200,000. Static pressure readings are recorded along the surface of the rotor test blade as well as a pressure traverse behind the stator blades. The second row of blades are movable with respect to the first row to simulate various positions to the wake interaction. The boundary layer growth is computed by the integral method presented in Part I and compared with another method presented in a recent NASA report. (NASA TN-D-5681 (McNally 1970).

TRANSPIRATION COOLING. PART III. USER'S MANUAL
Winget, L. E.; Han, L. S.
Ohio State Univ Research Foundation Columbus
Mar 73, 190p.
AFAPL-TR-73-11-Pt-3
AD-759 192
The computer program is presented for obtaining the drag and heat transfer coefficients around an airfoil with a blunt leading edge. The surface may be impermeable or porous with a specified blowing incorporating exponential profiles for both the velocity and temperature fields. The method has the capability of integrating around a body to the point at which the flow field separates from the surface. Provisions are included to handle the case of discontinuous blowing. In addition, any body for which the initial length can be approximated by the flat plate similarity solution can be analyzed.

TURBINE AND COMPRESSOR PERFORMANCE OF A BRAYTON ROTATING UNIT DURING HOT CLOSED-LOOP OPERATION
OPERATION
Wong, R. Y.
National Aeronautics and Space Administration. Lewis Research Center, Cleveland, Ohio
Sep 71, 27p.
NASA-TM-X-2350
N71-35232

THE ANALYTICAL TREATMENT OF SECONDARY FLOWS AND ASSOCIATED LOSSES IN AXIAL-FLOW TURBOMACHINES
Woods, J. R. Jr.
Naval Postgraduate School Monterey Calif.
10 Dec 71, 40p.
NPS-57W071121A
AD-734 983
The report presents the current state of the art of the analytical treatment of secondary flows and associated losses in axial-flow turbomachines. A physical description of the secondary-flow phenomena is given, as well as the theoretical methods that are available to treat secondary flows and to predict secondary losses.

AN INVESTIGATION OF SECONDARY-FLOW PHENOMENA AND ASSOCIATED LOSSES IN A HIGH-DEFLECTION TURBINE CASCADE
Woods, J. R. Jr.
Naval Postgraduate School Monterey Calif
Sep 72, 122p.
AD-750 183
The report presents precise quantitative data established for the overall flow losses in a high-deflection turbine rotor cascade for a range of aspect ratios h/c from 1.480 to 0.592. The magnitude of the losses due to secondary-flow effects was also determined. From these measurements it was possible to obtain an improved physical description of the complex secondary-flow phenomena.

HIGH-TIP-SPEED, LOW-LOADING TRANSONIC FAN STAGE. PART I. AERODYNAMIC AND MECHANICAL DESIGN.
Wright, L. C.; Vitale, N. G.; Ware, T. C.; Erwin, J. R.
Airesearch Mfg. Co., Los Angeles, Calif.
Apr 73, 180p.
NASA-CR-121095; AIRESEARCH-72-8421-PT-1
N73-22727

INLET FLOW FIELD SIMULATION TECHNIQUES FOR ENGINE/COMPRESSOR TESTING
Younghans, J. L.; Moore, M. T.; Collins, T. P.; Direnzi, J. G.
General Electric Co., Evendale, Ohio
Aircr Eng V 42, N 11, Nov 1970, p.12-17

A PREDICTION MODEL FOR LIFT-FAN SIMULATOR PERFORMANCE
Yuska, J. A.
National Aeronautics and Space Administration. Lewis Research Center, Cleveland, Ohio.
Aug 72, 71p.
NASA-TM-X-68788
N73-10282
The performance characteristics of a model VTOL lift-fan simulator installed in a two-dimensional wing are presented. The lift-fan simulator consisted of a 15-inch diameter fan driven by a turbine contained in the fan hub. The performance of the lift-fan simulator was measured in two ways: (1) the calculated momentum thrust of the fan and turbine (total thrust loading), and (2) the axial-force measured on a load cell force balance (axial-force loading), tests were conducted over a wide range of crossflow velocities, corrected tip speeds, and wing angle of attack. A prediction modeling technique was developed to help in analyzing the performance characteristics of lift-fan simulators. A multiple linear regression analysis technique is presented which calculates prediction model equations for the dependent variables.

PREDICTION OF GAS TURBINE ENGINE COMPRESSOR ROTOR BLADE SERVICE LIFE.
Zhuchenko, E. I.; Fridlender, I. G.
Sov Aeronaut V 16, N 4, 1973, p.100-103

ON THE PRINCIPLE OF REPEATABILITY AND ITS APPLICATION IN ANALYSIS OF TURBINE AND PUMP IMPELLERS.
Zienkiewicz, O. C.; Scott, F. C.
Univ of Wales, Swansea
Int J Numer Methods Eng V 4, N 3, May-Jun 1972, p.445-450

EFFECT OF PROJECTIONS ON THE SURFACES OF GAS TURBINE PARTS ON THE EFFICIENCY OF THEIR COOLING BY AN AIR FILM
Zolotogorov, M. S.
Teploenergetika N 12, Dec 1973, p.37-39

STUDY OF FILM-COOLING EFFECTIVENESS OF SOME GAS-TUBINE STATOR SURFACES
Zysin, V. A.; Zolotogorov, M. S.
Heat Transfer-Sov Res V 4, N 3, May-Jun 1972, p.6-10

POWER PLANT CONTROLS FOR AERO-GAS TURBINE ENGINES
Advisory Group for Aerospace Research and Development. Paris (France)
Mar 75, 374p.
AGARD-CP-151
N75-23575
Control requirements, control simulation techniques, and control system hardware for improved reliability of aircraft gas turbine engines are elaborated.

BEHAVIOR OF TURBOJETS
Deutsche Gesellschaft fuer Luft- und Raumfahrt, Cologne (West Germany)
Jan 73, 114p.
DIR-MITT-73-05
N73-33755
The influence of combustion chambers, compressors, and afterburners, and their mechanical configuration, on the static performance of two-cycle turbojet engines, with a high bypass ratio was investigated. Some examples are given to show the influence of the variation of both nozzle surfaces and fuel throughput at given thrust on the parameters of the same engines without primary and secondary flow mixing. The performance of arbitrarily switched gas turbines was calculated by simulation with building block system on a digital computer. The effects of a partly oil-filled converter for a turbine engine used to supply starting power to the main engines were investigated.

REPORT DOCUMENTATION PAGE			
1. Recipient's Reference	2. Originator's Reference AGARD-LS-83	3. Further Reference ISBN 92-835-1219-7	4. Security Classification of Document UNCLASSIFIED
5. Originator	Advisory Group for Aerospace Research and Development, North Atlantic Treaty Organization 7 rue Ancelle, 92200 Neuilly sur Seine, France		
6. Title	MODERN PREDICTION METHODS FOR TURBOMACHINE PERFORMANCE		
7. Presented at	Munich, Germany on 14–15 June 1976 and in London on 17–18 June 1976		
8. Author(s)	Various		9. Date June 1976
10. Author's Address	Various		11. Pages 170
12. Distribution Statement	This document is distributed in accordance with AGARD policies and regulations, which are outlined on the Outside Back Covers of all AGARD publications.		
13. Keywords/Descriptors	Turbomachinery Rotor blades (turbomachinery) Gas turbine engines Compressors Performance		14. UDC 621–135:621.438

15. Abstract

This Lecture Series, No.83 on the subject of Modern Prediction Methods for Turbomachine Performance, is sponsored by the Propulsion and Energetics Panel of AGARD and implemented by the Consultant and Exchange Programme.

Propulsion system development costs may be significantly reduced by improvement of methods for prediction of compressor and turbine component performance, and by preliminary study of the interactive operation of compressors and turbines with other system components. After the build-up of development engines, it is necessary to understand and carefully plan the process of rematching of components for optimum system performance.

AGARD Lecture Series Number 83 includes lectures and a panel discussion on the historical background of turbomachine performance prediction, on current procedures for estimation of overall and blade row performance characteristics, and on qualitative and quantitative turbomachine performance information needed for evaulation of the effects of compressor and turbine behaviour on the complete propulsion system. The lectures on component performance prediction cover both current and developing technology for axial-flow compressors and turbines, centrifugal compressors and radial-inflow turbines.

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ISBN 92-835-1219-7

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Printed by Technical Editing and Reproduction Ltd
Harford House, 7–9 Charlotte St, London W1P 1HD